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by

HAROLD ARTHUR MOORE, LIEUTENANT, U. S. NAVY
B. S., UNITED STATES NAVAL ACADEMY
(1956)

and

ANDERS TIMM ANDERSON, LIEUTENTANT, U. S. NAVY
B. S., UNITED STATES NAVAL ACADEMY
(1957)

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NAVAL ENGINEER AND MASTER OF SCIENCE IN NAVAL ARCHITECTURE
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Signatures of Authors. . . . .

. . . .
Department of Naval Architecture and
Marine Engineering, May 21, 1965

Certified by . . . .
Thesis Supervisor

Accepted by . . . .
Chairman, Departmental Committee on Graduate Students

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Thesis
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DESIGN CONSIDERATIONS FOR A SENSITIVE PROPULSION DYNAMOMETER FOR SMALL SHIP MODELS

by

LT. HAROLD ARTHUR MOORE, USN, and LT. ANDERS TIMM ANDERSON, USN

Submitted to the Department of Naval Architecture and Marine Engineering on May 21, 1965, in partial fulfillment of the requirements for the Master of Science Degree in Naval Architecture and Marine Engineering and the Professional degree, Naval Engineer.

ABSTRACT

The objectives of this thesis are to clearly define the problem of developing a dynamometer suitable for use in a small self-propelled ship model at the M.I.T. Towing Tank Facility, to explore the theories for the solution of the instrumentation problem and conduct the necessary preliminary design of promising methods, and to present design considerations for the detailed design, construction, and experimental phases of the problem.

The objectives are accomplished by determining the magnitudes of torque, thrust, shaft rotational velocity, and ship model velocity to be measured, by analyzing the anticipated disturbances, and by synthesizing and critically evaluating the measurement system components.

A design procedure for the stern tube bearing and shafting system is developed which produces constant torque losses. It is found that techniques at or near the state of the art must be employed to effect measurement of the desired alternating forces. Photoelectric, electron tube, piezoelectric, and variable capacitance transducer techniques standout as the most promising methods of measurement.

On the basis of practicality, it is recommended that a sensitive propulsion dynamometer capable of measuring only the mean values of torque and thrust be successfully developed before consideration be given to the development of a dynamometer capable of measuring the alternating forces.

Thesis Supervisor: Martin A. Abkowitz
Title: Professor of Naval Architecture

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I. INTRODUCTION

A. Motivation for Study

A dynamometer is an instrument for measuring force or power. A "sensitive propulsion dynamometer," for purposes of this thesis, is defined as a dynamometer capable of measuring the dynamic values of forces developed by the rotating propeller of a "small ship model" propulsion system. Specifically, these measurements are the dynamic values of torque, thrust, shaft rotational velocity, and ship model velocity.

The Massachusetts Institute of Technology Towing Tank facility does not presently include a propulsion dynamometer; therefore, the development and construction of a suitable dynamometer is desired. The size of the towing tank limits models used to a maximum length of five or six feet. The "small ship model," therefore, is defined for purposes of this thesis as a five or six foot scale model of a full-sized, single-screw surface ship.

The uses of the desired measurements may be placed under three general headings -- self-propulsion tests, evaluation of alternating force theory, and ship vibration studies.

Self-propulsion tests are conducted, using a model with known bare-hull resistance and developing thrust with a propeller of known characteristics, in order to determine the wake and hull interaction characteristics of the model. For these tests the mean values of torque, thrust, and shaft rotational velocity are required. A self-propulsion test would enable additional student participation in towing tank experiments and would provide the means for further investigation of scale effects in small, self-propelled models (1,2,3).

Several theories predicting alternating propeller forces have been developed and are discussed in references (4) and (5). The measurement of the instantaneous values of these forces will permit correlation between theoretical predictions and experimental data.

The propeller induced forces measured by the dynamometer are the driving functions required for solution of the ship and shaft equations of motion. Studies of ship and shaft vibrations made possible by the solution to these equations enable development of improved design of shipboard equipment and structures such as bearings and rotating components.

B. Background

1. Development trends - large ship models.

Through the years continuing programs of design and

development of propeller dynamometer instrumentation have been conducted at prominent model basin facilities throughout the world. Representative of these programs is the one which has been conducted at the United States Navy's David Taylor Model Basin (6). Here the instrumentation has been applied to models usually of twenty feet or of at least ten feet in length. The forces to be measured are more than thirty times those developed in the small ship model. As each new design evolved from the program great advances were made in overcoming the difficulties of obtaining a proper frequency response, a high signal-to-noise ratio, and a proper method of dynamic calibration. The latest of these designs has been very successful. The development of a pre-calibrated alternating force calibration, along with instrumentation which uses semi-conductor strain gage bridges practically on the propeller itself and amplification on the shaft prior to removal by slip rings, has been responsible for this success. This dynamometer is compact, versatile, and capable of measuring the six components of the propeller alternating forces (7). The DTMB report describing this dynamometer has not yet been published. A discussion of the applicability of these methods to the small ship model will be given in later sections of this thesis.

2. Historical development - small ship models.

Several attempts at designing and testing a dynamometer for measuring the mean and/or alternating values of torque and thrust developed by small ship models have been made as thesis projects at M.I.T. (8,9,10,11,12).

The schemes of four of these projects dealt principally with the use of strain gages for torque sensing and a differential transformer for thrust measurements. These devices converted physical strains and displacements into electrical signals which were removed from the shaft by a slip ring assembly. The most significant result of the four projects was a distinct, although still inadequate, progress in the development of slip rings. Development progressed from a copper ring and relay contact brush assembly (8) to a simply constructed mercury pool slip ring construction (11). No satisfactory results, however, were obtained in the testing of these schemes. The system natural frequencies were too low for an undistorted detection of the alternating forces; and the signal-to-noise ratios were too low for a distinguishable output. The most apparent reason for the failure was the authors' failure to adequately analyze the details of the problem with which they

were faced. Whether the goal was to measure the alternating forces or only the mean values, a proper analysis of the forces themselves, the sensitivities and devices required for detection, and the tolerances for construction would have enabled some of the failures to be predicted rather than revealed through trial and error.

The frequency requirements for measurement of the alternating forces were recognized in the fifth of these theses (12), and considerable emphasis was placed on the frequency response in defining the problem. Having treated this aspect, the authors directed their efforts to the solution of the instrumentation problem through construction and testing of a suggested device. The variable reluctance principle was used to detect thrust actuated variations in a small air gap. Although the application of the principle as an instrumentation technique was presented very well, the sensitivity and resolution required to effect the desired measurements were not evaluated. Construction difficulties prevented complete testing. Had complete test results been obtained, perhaps the required resolution might also have been obtained as an extra bonus. But, as in the other theses, a more detailed analysis of the

problem might also have enabled prediction of some of the failures.

C. Objectives

Instrumentation engineering requires proper developments of the theory, preliminary design, detailed design, construction, and experimentation of the systems that produce functional relationships among physical quantities. Originally, the authors of this thesis expected to embark on a program of design, construction and experimentation similar to those already conducted. A careful study of the past theses, however, indicated that greater emphasis was required in the development of the theory and preliminary design. This development, along with sufficient economic backing to ensure proper construction and availability of required equipment, is necessary to ensure successful construction and experimentation. Therefore, the authors established the following new objectives:

1) To clearly define the problem of developing a dynamometer suitable for use in a small self-propelled ship model at the M.I.T. Towing Tank Facility.
2) To explore the theories for the solution of the instrumentation problem and conduct the necessary preliminary design of promising methods.

3) To present design considerations for the detailed design, construction, and experimental phases of the problem.

D. General Organization

This thesis is organized into four major sections. The method by which the thesis objectives will be accomplished is presented in the Procedure section. The measurement system, with its inputs and output, is analyzed in the Analysis section. The Analysis is followed by the Conclusions and Recommendations.

II. PROCEDURE

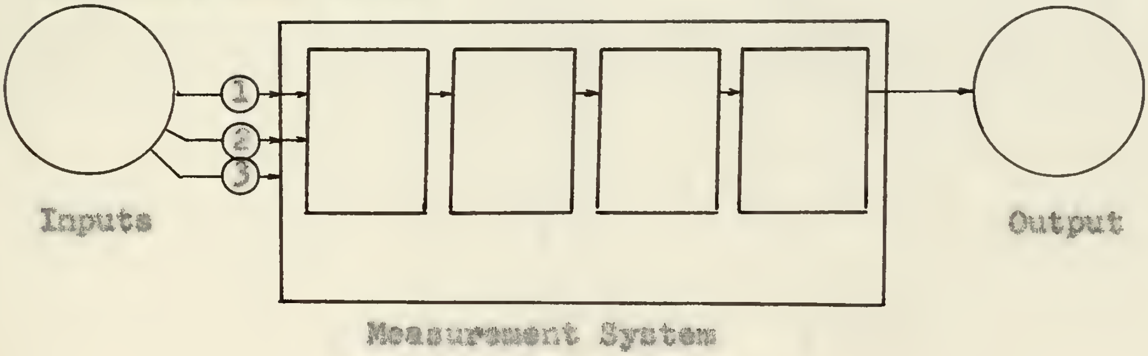
A. Analytical Procedure

The objectives of this thesis will be accomplished by the following analytical procedure:

1. Analyze the primary or desired forces, i.e. determine the magnitudes of torque, thrust, shaft rotational velocity, and ship model velocity.
2. Analyze the anticipated disturbances.
3. Synthesize and critically evaluate the measurement system components, presenting design considerations for each.

This procedure requires the formulation of an analytical model.

B. Analytical Model



1. Inputs

Branch ① - Desired inputs to the measurement system are axial propeller thrust, propeller torque, propeller shaft rotational velocity, and ship model velocity.

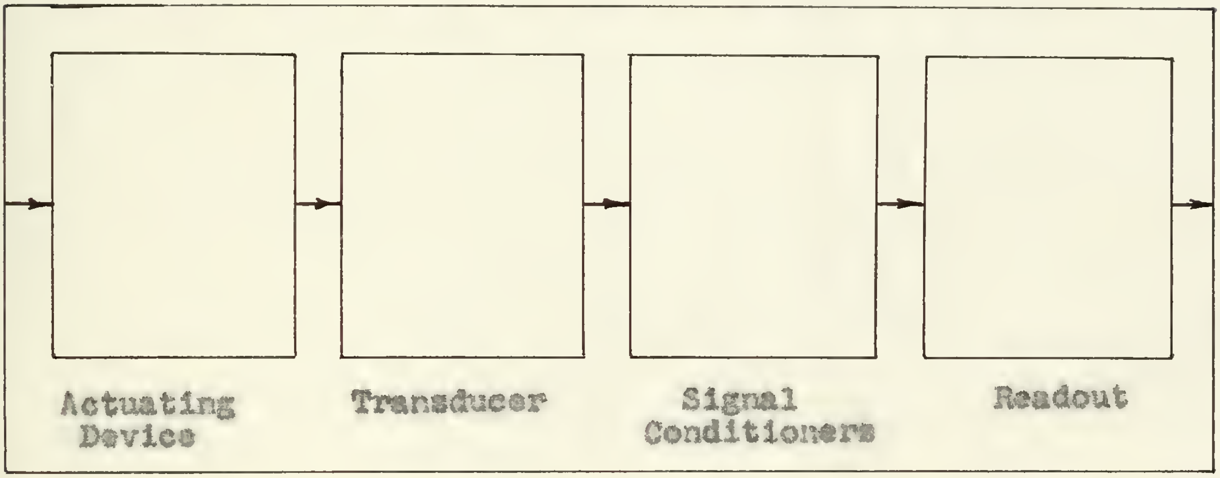
Undesirable inputs, referred to as disturbances or noise, are all other forces and effects that enter the measurement system and are capable of masking or mixing with the desired input.

Branch ② - Noises which can enter the measurement system in the same manner as the desired inputs are bearing forces, prime mover noise, disturbances acting on the propeller, and forces created by the measurement system itself.

Branch ③ - Noise can mix with the desired inputs in the various elements or components of the measurement system. Examples of these noises are: spurious signals resulting from electrical parameter variations due to temperature or humidity changes, slipring noise, electrical and magnetic field influences, component distortion due to centrifugal force, and power supply variations.

2. Measurement System

The measurement system can be defined as that system which detects the physical quantities and converts them to a suitable output data form. The analytical model of the measurement system chosen is that of the American Standards Association (13) and is described as follows:



a. Actuating device - that device or mechanism that converts torque and thrust into the measurable quantities strain, displacement, velocity, or acceleration. Types of actuating devices considered are: bellows, diaphragms, shaft rotations, rectilinear motions, proving rings, cantilever beams, seismic mass and springs.

b. Transducer - the element that converts the measurable quantity to a signal. Types of transducer elements considered are self-generating analog, variable parameter analog, frequency or pulse generating, and digital.

c. Signal conditioners - those devices that condition the transducer output signal and prepare it for actuating the readout device. Types of signal conditioners considered are input modification devices, instrumentation amplifiers, and signal removal methods.

d. Readout - those devices that accept the conditioned signals and present them in a suitable output data form. Types of readout devices considered are analog or digital indicators and recorders.

3. Output

Output data can be analyzed in many ways. The input characteristics and data analysis techniques used largely determine the desired form of this data. Output data can be in the form of visual display, strip charts, tape recordings, punched paper tape, and punched cards.

III. ANALYSIS

A. General

The development of a dynamometer as an instrument involves a compromise of four fundamental qualities: efficiency, scale range, sensitivity, and accuracy (14). The efficiency of an instrument is a measure of the amount of disturbance which it introduces into the system. The scale range is the difference between the largest and the smallest possible readings. The sensitivity is the slope of the instrument calibration curve, that is, the ratio of the change in measured output to a given change in input. The accuracy is the maximum error -- the difference between the indicated quantity and the true quantity. If desired as a percentage, accuracy can be expressed in two ways:

$$\frac{\left|Q_{\text{indicated}} - Q_{\text{true}}\right|_{\text{max.}}}{Q_{\text{true}}} \times 100\ \text{o/o}$$

$$\frac{\left|Q_{\text{indicated}} - Q_{\text{true}}\right|_{\text{max.}}}{Q_{\text{full scale}}} \times 100\ \text{o/o}\ ,$$

where Q represents the quantity of interest and full scale pertains to the maximum anticipated amplitude of the quantity of interest. If this expression for accuracy is applied to

a meter device, then the maximum amplitude is the maximum meter deflection. Resolution, a term often used in conjunction with accuracy, is the degree to which nearly equal values of a given quantity can be discriminated. The higher the value of resolution stated, the more minute is the discernible difference between values.

The above qualities can be applied to the measurement system as a whole, or to the system components individually. Since the qualities of the components determine the quality of the system, they will be considered in the analysis of the individual components.

As described in the Procedure section, the measurement system is composed of the transducer section, the signal conditioning section, and the readout section. The analyses of the inputs to this system, the system itself, and its output follow.

B. Inputs

1. Discussion

There are three general categories of forces -- static, transient, and dynamic. Static forces are steady and not subject to rapid fluctuations or discontinuities. Transient forces result from a change in operating conditions, such as speed changes or rudder angle changes, and last only a short period of time. Dynamic forces are continuous and either periodic or random in nature.

When a propeller is operated in an unsymmetrical wake behind a ship, each blade experiences periodic wake variations which induce periodic forces in the propeller shafting system. These periodic forces appear at frequencies related to the multiples of shaft rotational frequency. In addition to these periodic forces, random forces are produced by the turbulence in the flow around the propeller. Thus, the desired propeller-induced forces are dynamic in nature and neither transient nor static. The dynamometer, therefore, is to be designed for measurement of dynamic forces.

In order to enable the authors to proceed through the analysis with specific input values, only the forces induced by the wake variations of such ship models as the Mariner Class and Series 60 will be discussed. These models are typical and currently available for use in the M.I.T. towing tank.

2. Desired quantities

a. Torque and Thrust

The forces induced in the shafting system by a propeller operating in an unsymmetrical wake result in axial thrust, torque, and shaft bending moments. In this thesis, only the axial thrust and shaft torque are considered to be desired

inputs to the measurement system. In addition, ship model velocity and shaft rotational velocity are required for calculation of effective horsepower and shaft horsepower respectively.

Table I presents the anticipated range of mean values of the desired forces and velocities of the small merchant-type models to be tested. The computations for the tabulated values are given in Appendix A.

Table I

Anticipated Range of Forces and Velocities of Small Merchant-type Models

Ship	V_{ship} (kts.)	V_{model} (kts.)	$Torque_m$ (in.-oz.)	$Thrust_m$ (lbs.)	RPM_m	RPS_m
Mariner	14	1.4	0.77	--	628	10.5
	21	2.1	2.24	--	998	16.6
Series 60	10	0.914	0.25	0.047	444	7.4
	22	2.01	1.75	0.203	1130	18.8
Tanker	10.9	0.992	--	0.101	--	--
	18.5	1.688	--	0.315	--	--

The following specific values are based on the information presented in Table I and were selected for the design analysis of the dynamometer as the full scale mean values for the highest velocity test run:

full scale mean torque (highest velocity of interest)

= 2.30 in.-oz.

full scale mean thrust (highest velocity of interest)

= 0.32 lb.

The full scale mean values for the test run conducted at the lowest velocity of interest, 65% of the highest velocity, were appropriately scaled from the values above and are as follows:

full scale mean torque (lowest velocity of interest)

0.970 in.-oz.

full scale mean thrust (lowest velocity of interest)

0.135 lb.

The smallest values of the periodic, propeller-induced torque and thrust must be determined before the instrument scale range and the resolution required for their measurement can be established. The magnitudes of the torque and thrust variations are functions of four factors:

1) The shape of the afterbody.
2) The clearance between the propeller and and the screw aperture.
3) The number of blades and the hydrodynamic properties of the propeller blades.
4) The ship operating conditions.

A discussion of some of these factors and a method for estimating the magnitudes of the torque and thrust variations are presented in Appendix A. The following specific values are based on the method presented and represent the smallest variation amplitude of interest, that is, the variation amplitude for the test trial conducted at the lowest velocity of interest:

smallest amplitude of torque variations
= 0.004 x 2.30 in.-oz.= 0.0092 in.-oz.
smallest amplitude of thrust variations
= 0.01 x 0.32 lb. = 0.0032 lb.

The David Taylor Model Basin stated in reference (6) that their design goal for accuracy is 0.25 o/o of the full scale of interest. To adopt this accuracy as a standard for design of the dynamometer under consideration, means that the

following values represent the maximum tolerable error:

$$\left| Q_{\text{indicated}} - Q_{\text{true}} \right|_{\max} = \frac{Q_{\text{full scale of interest}} \times 0.25\ \text{o/o}}{100}$$

$$= 0.0092\ \text{in.-oz.} \times 0.0025$$

$$= 0.000023\ \text{in.-oz. of torque}$$

and

$$= 0.0032\text{lb.} \times 0.0025$$

$$= 0.0000080\ \text{lb. of thrust}$$

These values of maximum tolerable error, therefore, establish the lowest acceptable resolution which is compatible with the desired accuracy; the dynamometer should at least be capable of discriminating between values separated by these magnitudes. Such resolution may be stated as being one part in four hundred based on the full scale of interest.

The gravity dynamometer currently used at the M.I.T. Towing Tank for measuring the bare-hull resistance of small ship models is accurate to 0.0001 lb. (15). Since the gravity dynamometer will be used in conjunction with the propulsion dynamometer for self-propulsion tests, the accuracies of the two measurements should be compatible. The full scale mean values for the lowest velocity of interest are the values of interest for comparison. The maximum tolerable error, if the dynamometer was designed only for mean value measurements

at the adopted standard of accuracy, would be as follows:

$$\left| Q_{\text{indicated}} - Q_{\text{true}} \right|_{\max} = 0.970 \text{ in.-oz.} \times 0.0025$$

$$= 0.00243 \text{ in.-oz. of torque}$$

and $$= 0.135 \text{ lb.} \times 0.0025$$

$$= 0.00034 \text{ lb. of thrust.}$$

Thus the accuracies for the thrust measurements are not compatible. Because the propulsion dynamometer is being designed to measure both the variations and the mean, however, the comparison of 0.0001 lb. to 0.000008 lb. of thrust shows the design accuracy of the propulsion dynamometer to be more than sufficient.

It is recognized that, in order to attain the accuracy adopted as a standard for design, the maximum tolerable error values are very small. Anticipating the requirement for a possible relaxation of the accuracy standard, the maximum tolerable errors for compromise accuracies have been computed and are plotted in Figure 1.

Knowledge of the range of frequencies of interest for the torque and thrust variations is as important to the analysis of the measurement system as the information regarding the magnitudes. The range of frequencies of interest is shown

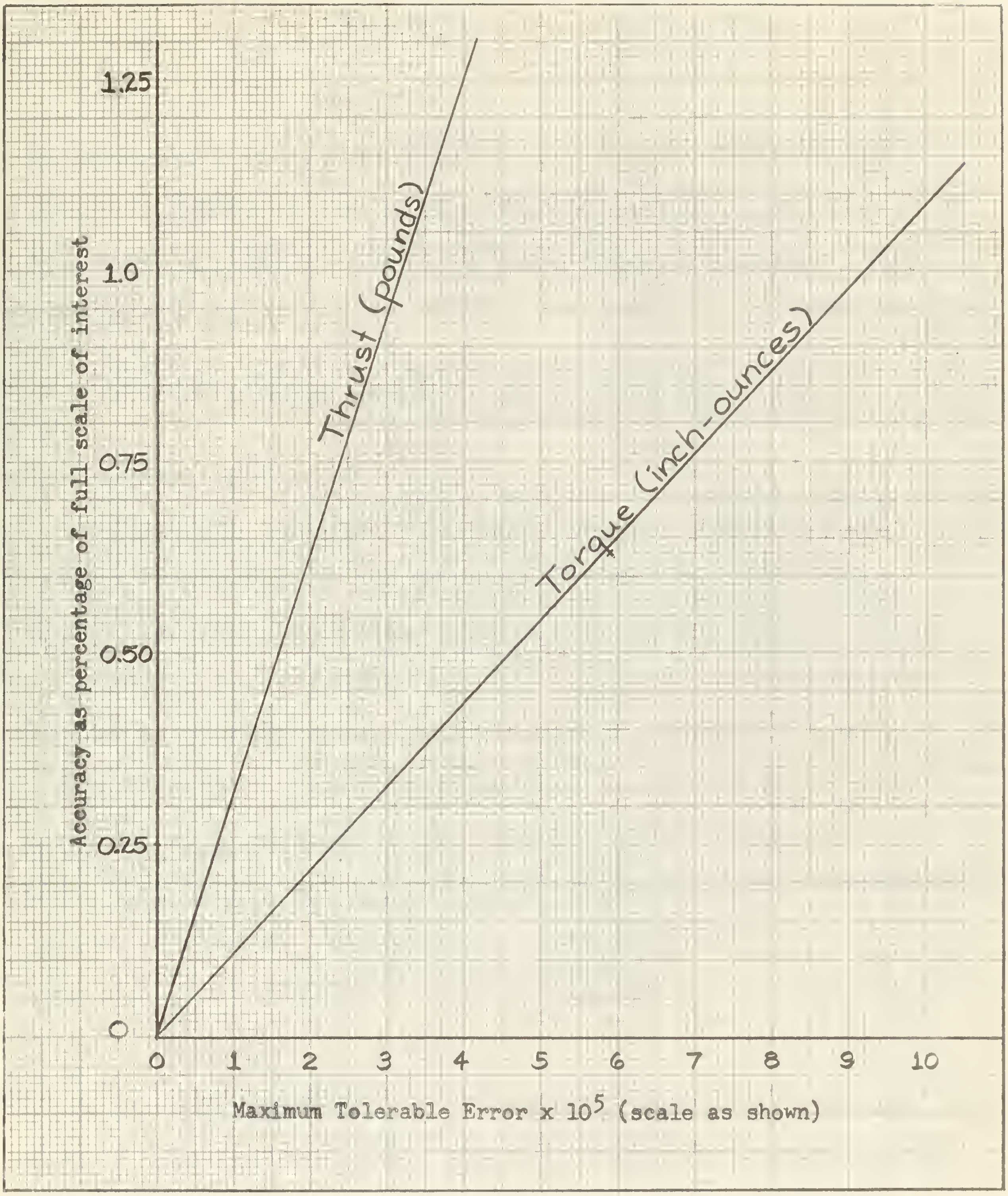


Figure 1. Relationship Between Accuracy and Maximum Tolerable Error in the Measurement of Torque and Thrust Variations.

in Appendix A to extend from the shaft rate (RPS) to the second harmonic of the blade rate for a five-bladed propeller (2 x 5 RPS). Based on the highest full scale shaft rotational velocity, the highest frequency of interest is:

$$2 \times 5 \times 20 \text{ RPS} = 200 \text{ cycles per second}$$

b. Velocities

Since the constant velocities, both rotational and linear, are required for the tests to be conducted using the propulsion dynamometer, the accuracy required for their measurement is dependent upon the capability of the driving device to provide the constant velocity. Linear velocity can presently be measured at the M.I.T. Towing Tank to an accuracy of 0.001 knot (15). The accuracy required for the shaft rotational velocity is determined by the frequency of velocity input necessary for successful operation of a constant speed control device for the prime mover.

c. Summary

The full scale values of interest for torque, thrust, shaft rotational velocity, and model velocity are summarized in Table II for the lowest and highest test velocities of interest.

Table II

Selected Input Forces, Velocities, and Frequencies for Measurement System Analysis

Percentage of Full Speed		65 o/o	100 o/o
Model velocity (kts.)		1.3	2.0
Mean values	Torque (in.-oz.)	0.970	2.30
	Thrust (lb.)	0.135	0.32
Smallest variation amplitude of interest (2^{nd} harmonic, 5 bladed propeller)	Torque (in.-oz.)	0.0092	0.016
	Thrust (lb.)	0.0032	0.0058
Shaft rotational velocity (RPM/RPS)		780/13	1200/20
Highest frequency of interest (cps)	3 blades 2nd harmonic	78	120
	4 blades 1st harmonic	52	80
	5 blades 2nd harmonic	130	200

3. Disturbances or Noise

The description, analysis, and means of suppression of those disturbances which enter the measurement system in the same manner as the desired inputs will be presented in this

section. The disturbances which can mix with the desired inputs in the various elements of the measurement system will be discussed individually when they affect the individual components.

In order for disturbances to enter the measurement system in the same manner as the desired inputs, they must be either originated on the shaft or imparted to the shaft. The disturbances which originate on the shaft may be initiated by field force disturbances within the measurement system itself or by the axial component of the gravity force on the shafting system. Disturbances which are imparted to the shaft may be transmitted through the propeller, the prime mover, and the bearings.

a. Propeller

In addition to the propeller-induced forces producing torque and thrust, four other components are present -- the horizontal and vertical transverse forces and the horizontal and vertical bending moments. These four components represent unwanted quantities to the measurement system, but, since they act in directions which are different from those of the desired quantities, the measurement system can be designed to neglect them. These components will, however, affect the loadings in the stern tube bearing where they may be transmitted back to the shaft as components in the desired directions.

Force variations in the desired directions are also induced by flexure motions excited when forces are applied to the shaft. These additional thrust and torque variations are termed propeller added mass and damping, and result from a certain amount of axial and rotational displacement of the propeller. In order to eliminate the added mass and damping, and thereby limit the propeller response to only the excitation forces desired, the shaft spring constant must be high. Unwanted wave disturbances also may be transmitted to the shaft through the propeller, but the elimination of these disturbances is a problem of creating the desired tank conditions prior to testing.

b. Prime mover

Disturbances generated by the prime mover may be either imparted directly to the shaft or transmitted via the structureborne path -- prime mover foundation, hull, bearing, shaft. An electric motor is best suited for the prime mover in the small ship model; and the magnetic forces, acting across the air gap of an electric motor, are a major source of disturbances. These forces are pulsating due to variations in the magnetic path as the rotor rotates and the electric load currents change. These pulsations

then set up vibrations in the shaft and foundation. Other forces originating in the magnetic structure such as magnetostriction, eddy currents, and hysteresis are not significant factors (16).

The goal of this thesis requires that prime mover disturbances be eliminated. The shaft vibrations due to the magnetic forces can be damped by a suitable shaft flywheel and flexible coupling arrangement placed between the prime mover and the measurement system. The vibration path through the structure can be blocked by appropriate isolation mounting of the motor foundation. In addition to isolation methods, proper selection of the number of poles and field excitation frequency can place the motor disturbance frequencies outside the frequency range of interest for the desired quantities.

c. Bearings

The bearings align and support the shaft between the prime mover and the propeller. The disturbances transmitted to the shaft are caused by the friction which opposes the turning of the shaft in the bearing. The value of the friction depends on the method of lubrication, the lubricant, its temperature, the velocity of rotation, and the

intensity of pressure on the bearing. The intensity of pressure varies with the relative motion between the shaft system and the bearings, regardless of its origin. These motions can be produced by shaft system and hull vibrations, shaft misalignment, and shaft loading.

Proper bearing selection is the first step to suppression of bearing noise. The design and application of bearings in low noise equipment have been studied extensively (16), and as a result of these studies, the following comparisons between ball bearings and sleeve bearings can be made:

1) Irregularities in the bearing balls and races and internal end play in the bearings themselves are sources of noise in ball bearings. Closely controlled manufacture of ball bearings, however, results in uniformity in performance and permits the maintenance of air gap concentricity in motors and instrumentation throughout the life of the bearing. Considerable study of the internal geometry of ball bearings has resulted in their availability under specifications

which have very low noise levels. Other factors which affect the noise level of the ball bearings as installed in the equipment are the amount of axial preloading, the alignment on the shaft, and the type and cleanliness of the lubricant.

2) Sleeve bearings are inherently quieter than ball bearings but there are also certain disadvantages in their application. Sleeve bearings require a radial clearance which incr$ases with wear. This wear may result in eccentric air gaps in motors and instrumentation thereby creating unbalanced magnetic forces. The lubrication system used for the sleeve bearings can also be a source of noise.

3) Specific aspects of motor structureborne noise were also investigated. A motor which could use either sleeve bearings or ball bearings was tested extensively and it was found that the overall noise levels from

either the ball or sleeve bearings were about the same. The overall levels for the sleeve bearings, however, were made up primarily of lower frequency vibrations and those for the ball bearings were made up primarily of higher frequency vibrations. All factors considered, the ball bearings were preferable to sleeve bearings for use in motors.

In selecting bearings for the small ship model, several points must be considered. Wear is not a problem since testing time will be short and bearing loadings will be light. The presence of water at the stern tube creates a problem for ball bearing lubrication but provides a natural lubricant for a sleeve bearing. Model stern tube structure favors the incorporation of a sleeve bearing with a shaft seal configuration. Several possible instrumentation configurations favor the use of ball bearings in the shaft and motor. After considering these points, the authors decided to analyze a shaft-bearing configuration consisting of a sleeve type stern tube bearing and ball type shaft and motor bearings.

In general, past small ship model investigations (8,9,10,11,12) were limited either by the assumption that torque losses through the stern tube bearing were of the same order of magnitude as the torque to be measured, thereby requiring placement of the sensing elements between the bearing and the propeller; or by conclusions drawn from calculations which showed the torque loss to be negligible compared to the full load torque, thereby allowing placement of the sensing elements inside the model just forward of the stern tube bearing. Neither of these choices properly describes the difficulties which the bearing imposes on the problem of effecting the desired measurements. Regardless of the placement of the sensing elements, forward or aft of the stern tube bearing, it is the variations in torque loss that will affect the measurement of the desired alternating torque. The torque loss variations must be shown to be negligible compared to the alternations to be measured. A constant torque loss can be removed by proper dynamic calibration of the dynamometer.

Whereas every effort should be made to eliminate the motions which cause torque loss variations, it is impossible to remove them completely. A procedure for

design of the stern tube bearing and shaft for the small ship model has been developed using journal bearing analysis and design techniques (17) for a bearing length to shaft diameter ratio of one. This design procedure is presented in Appendix B and shows that a bearing can be designed for which the torque loss is constant over the operating range of the small ship model.

In regard to the installation of shaft and motor bearings, it is anticipated that certain instrumentation configurations and a possible slip ring installation might cause the slightest radial and axial play in the bearings to be intolerable. So-called "preloaded bearings" are used for such applications; but even these bearings, designed at the present state of the art, develop radial and axial play from wear and eccentricities. The discrete frequencies of ball bearing disturbances are in the frequency range of interest for measurement of the desired quantities. When shaft and bearing sizes are selected, the exact frequencies can be determined from information given in reference (18).

Information concerning the amount of play which might be expected in bearings applied to the small ship model was collected from the M.I.T. Instrumentation Laboratory. It

was learned that axial play in bearings designed especially for use in missile guidance gyros has a measured value in the vicinity of two to ten microinches (19). Radial play is expected to be approximately of the same magnitude. Work is presently in progress at the Laboratory to measure the actual play in a newer bearing design; values in the vicinity of one-fifth of a microinch for radial play and one-tenth of a microinch for axial play are expected. Although the actual cost information for such precision bearings was not available, it is suspected that if these bearings were required for the small ship model, the cost would be prohibitive.

After the shafting system is installed in the ship model some unbalances may still exist which cause excessive bearing loadings. An interesting method of continuous balancing of rotating parts is mentioned in reference (20) and is a scheme which could be applied to the small ship model if weight considerations will permit. The scheme consists of three steel balls placed inside and free to circulate within a circular rim which is nominally concentric with the rotating member to be balanced. When driven at high speeds, the balls quickly assume positions such that their dynamic effect nullifies the residual unbalance of the system.

C. Measurement System

1. Preliminary

a. Dynamometer categories

Having determined the desired dynamometer inputs, the next step in the design procedure is to select the proper method to measure these quantities. In general, dynamometers fall into three distinct categories: the power absorbing dynamometer, the driving dynamometer, and the transmission dynamometer.

1) The power absorbing dynamometer.

The power absorbing dynamometer is simply a device to absorb the output shaft power and to indicate the magnitude of the power it is absorbing. In the ship propulsion application, however, the power is required to propel the ship by dissipation of energy to the water. Since the towing tank represents such a large reservoir for the amount of energy dissipated, it would be impractical to measure the power being absorbed.

2) The driving dynamometer.

The distinguishing aspect of the driving dynamometer is that it provides both the mechanical power required and also

the measurement of the power. The electric cradled dynamometer is a d-c machine with the field structure cradled in trunion bearings. The assembly is free to rotate about the shaft except for the restraint provided by springs upon which the torque reactions are measured. This reaction type of driving dynamometer is promising for torque measurements when frequency requirements do not limit the shafting and motor weights.

A motor can be used as a driving dynamometer if its losses are known and the electric input is measured. The total loss can be determined as the sum of the rotation loss, armature-resistance loss, and brush loss. For precise measurements, the determination of the brush loss becomes a problem. The motor may be either an a-c induction or d-c motor. The calibration of an induction motor to determine losses, however, is considerably more difficult than that of a d-c motor but is practical if not too much precision is expected (21).

3) The transmission dynamometer.

The transmission dynamometer is the most versatile of the three types because it enables measurement of the torque

or thrust transmitted and neither adds to nor subtracts from the transmitted energy. A transmission torquemeter usually takes the form of a special coupling which connects the prime mover and the load without much shaft or machinery modification. A transmission thrustmeter takes a similar form connecting the propeller and the point of thrust transmission to the ship.

Since the power required to propel the ship is proportional to the product of the axial thrust developed by the propeller and the velocity of the ship, no direct power measurements can be made. The true thrust, therefore, must be measured by a transmission dynamometer.

Although the torque measurement could possibly be made with an electric cradled dynamometer, a d-c driving dynamometer, or a transmission dynamometer, fewer difficulties are encountered in the use of the transmission dynamometer. Emphasis will be placed on the transmission type in the analysis of the measurement system. Weight considerations and brush loss determination are the particular difficulties which make the reaction and driving dynamometers less desirable.

b. Seismic instruments versus fixed reference instruments.

In a seismic instrument, the base of a mass-spring system is attached at the point where the vibration is to be measured. The motion at the point is inferred from the motion of the mass relative to the base. In a fixed reference instrument, one terminal of the instrument is attached to a point that is fixed in space and the other terminal is attached (e.g. mechanically, electrically, optically, etc.) to the point whose motion is to be measured (22). The seismic device, as it would be utilized in this dynamometer, would result in two dynamic systems being coupled together, each with its own dynamic response characteristics. This results in the possible errors in the shafting dynamic system being detected and amplified in the seismic system. For this reason, a fixed reference instrument is selected for this dynamometer. The location of the fixed reference point remains to be determined.

2. Actuating device

a. Purpose

The purpose of the actuating device is to convert the input forces to the measurable quantities -- strain,

displacement, velocity, and acceleration. Strain is actually a form of displacement and will initially be treated as such in this development.

b. Equations of motion

The differential equation of motion for the single degree-of-freedom system with viscous damping, when excited by a force $F = F_o \sin\omega t$ (applied to the mass) is:

$$m\ddot{x} + c\dot{x} + kx = F_o \sin\omega t$$

where the force is either thrust or torque, and the displacement (at this point) can be either an axial or torsional displacement. The solution to this equation can be expressed in the form developed in reference (22):

$$x = \frac{R_d F_o \sin(\omega t - \theta)}{k}$$

The response factor for displacement, R_d, is the ratio of the amplitude of the vibratory displacement to the spring displacement that would occur if the force F were applied statically. R_d can be differentiated to produce velocity and acceleration response factors, R_v and R_a. These response factors are expressed as follows:

$$R_d = \frac{1}{\sqrt{(1 - \omega^2/\omega_n^2)^2 + (2\ c/c_c \cdot \omega/\omega_n)^2}}$$

$$R_v = R_d \cdot \omega/\omega_n$$

$$R_a = R_d \cdot \omega^2/\omega_n^2$$

The phase angle θ is expressed by:

$$\theta = \arctan \frac{2\ c/c_c \cdot \omega/\omega_n}{1 - (\omega/\omega_n)^2}$$

where: ω = excitation frequency

ω_n = system natural frequency

c = system damping

c_c = critical damping

Curves showing the dimensionless displacement factor R_d as a function of the frequency ratio ω/ω_n are plotted in Figure 2 on the coordinate lines having a positive 45° slope. This figure shows graphically the velocity and acceleration response factors, R_v and R_a, the former referred

to the horizontal coordinates and the latter referred to the coordinates having a negative 45° slope. Curves of the phase angle θ are plotted in Figure 3.

c. Selection of measurable quantities

An examination of Figures 2 and 3 reveals the following information pertinent to selection of a measurable quantity:

1) Displacement measurement requires a high ω_n (low ω/ω_n) to suppress harmonic amplitude and phase distortion.

2) Velocity measurement requires operation near resonance, requires high damping, does not treat all harmonics uniformly, and would require recalibration for each change in operating frequency ω.

3) Acceleration measurement requires a low ω_n to suppress harmonic amplitude and phase distortion.

Velocity measurement is the least desirable of the three quantities discussed and will be dropped from further consideration for the reasons given above. The high ω_n required for displacement measurement increases the allowable spring stiffness, for a given mass. This results in lower sensitivity but good rise time. On the other hand, the low ω_n required for acceleration measurement reduces the allowable spring stiffness to the point that the effect of added mass and

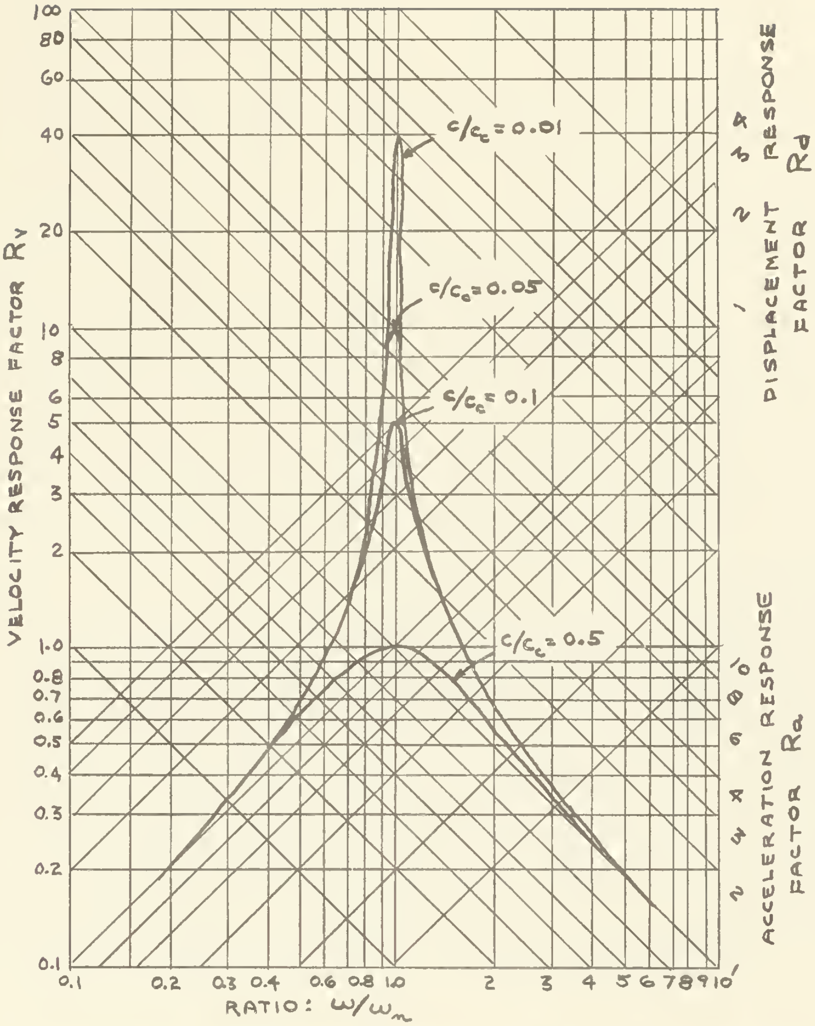


Figure 2. Response Factors for a Viscous-Damped Single Degree-of-Freedom System Excited in Forced Vibration

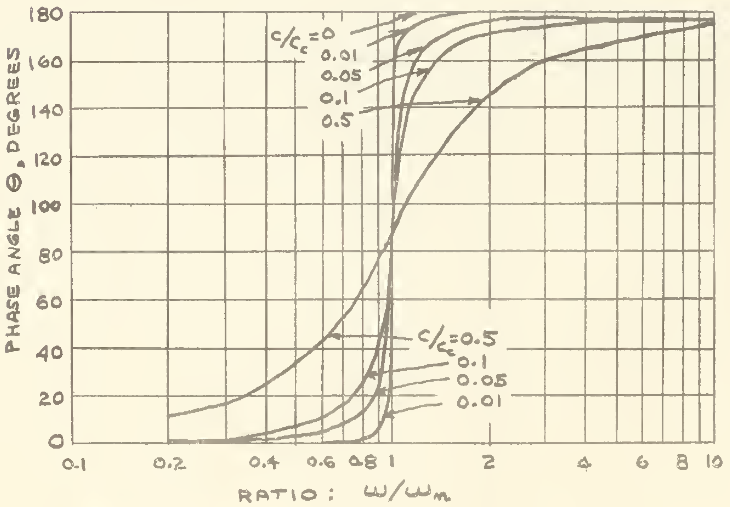


Figure 3. Phase Angle Between Response Displacement and Excitation Force for Single Degree-of-Freedom System with Viscous Damping

damping of the propeller, discussed earlier in this thesis, may become a noticeable, and undesirable effect. Therefore, the authors have elected to investigate only displacement measuring instruments.

d. Selection of measurement system natural frequency

Figures 2 and 3 indicate that, for any given periodic input force, the measurement system natural frequency, ω_n, and damping, c, are the two parameters that determine the system response. These figures show that, with the damping ratio $c/c_c = 0$, a value of ω/ω_n below approximately 0.10 is required to prevent magnification of higher harmonics, but that no phase distortion or shift occurs until $\omega/\omega_n = 1.0$; and that with c/c_c approximately 0.65, very little amplitude magnification occurs until ω/ω_n exceeds 0.4, but that phase shift increases approximately linearly with frequency ratio. Each harmonic of the input signal is retarded uniformly and retains the same relative harmonic relationship with the other terms; thus there is no phase distortion, but the entire response is shifted in time relative to the input signal.

The measurements of propeller damping shown in reference (23) indicate that the value of c/c_c for a propeller shafting system is low. A method for actual determination of propeller damping of a particular propeller is given in reference (24). For discussion purposes in this thesis, c/c_c will be assumed to be between 0.0 and 0.1. This damping ratio requires an instrument natural frequency of at least ten times that of the highest frequency of interest, or, for this design, 2000 cps.

e. Selection of measurement system configuration and determination of displacements.

The configurations of the propeller shafting and measurement systems determine the values of the mass and moment of inertia terms appearing in the various equations of basic vibration theory to be used. Having selected an actuating device natural frequency, a determination of the mass and moment of inertia essentially determines the allowable stiffness of the actuating device. The stiffness determines the magnitude of the measurable quantities, axial and torsional displacement.

Before the configurations can be selected, however, the position of a suitable fixed reference must be considered.

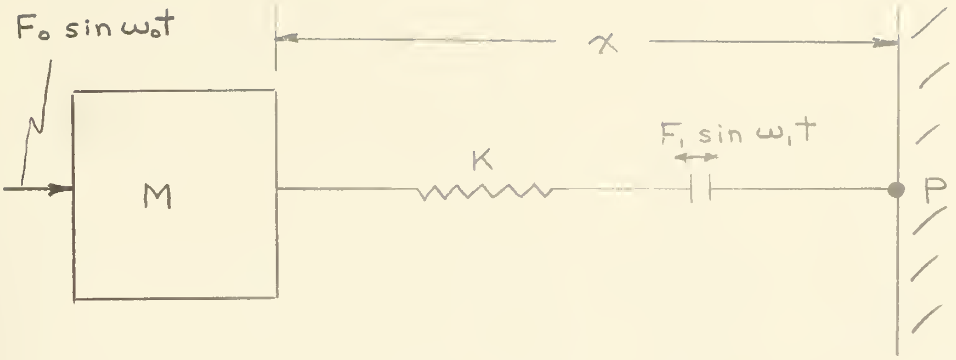
This consideration involves the possible interdependency of the propulsion system and the measurement system.

As F. M. Lewis points out in reference (25), the dependency of this point on the dynamic properties of the model must be made as small as possible. Ideally, the reference point must be located so that it has the least possible effect on the measurement. Two possible thrust measurement configurations are considered:

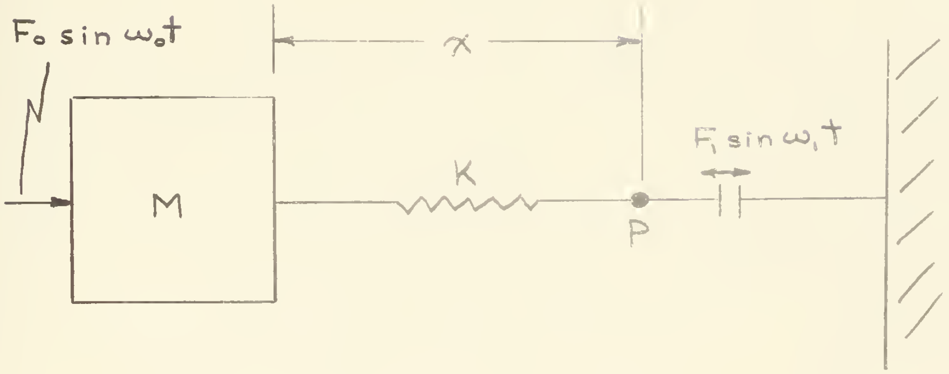
1) Measurement of the displacement of the propeller relative to the model hull considering the hull as the fixed reference point.
2) Measurement of the displacements of one end of the propeller shaft relative to the other end considering the end of the shaft nearest the thrust bearing as the fixed reference point.

These configurations will be analyzed with the assistance of Figure 4.

In Figure 4(a), the reference point is on the hull; in Figure 4(b), the reference point is on the propeller shaft forward of the measurement system actuating device. In both cases it is desired to make the measurement of



a) Reference point on ship hull



b) Reference point on propeller shaft

M = propeller and hub, entrained water, and some shafting
K = measurement system actuating device stiffness
F_o = desired input force magnitude
ω_o = excitation frequency of interest
F_1 = disturbance force from thrust bearing
ω_1 = frequency of thrust bearing force
x = measured displacement
P = measurement system reference point

Figure 4 Analytical Models of Propeller Shafting System and Ship Hull for Measurement Reference Point Determination

displacement, x, be an accurate measurement of the desired input force F_o. In theory, this could be done through proper selection of system parameters; however, the presence of a positive displacement in the thrust bearing influences the measured displacement. This influence can be analyzed by considering the case of a mass-spring system with excitation applied to the end of the spring. As shown by Den Hartog in reference (26), the disturbance does not change the spring length when the bearing disturbance frequency, ω_1, is much lower than the natural frequency of the mass-spring system. Thus, the bearing disturbance "sees" the mass and spring as a stiff rod. Since the bearing disturbance is one of positive displacement, motion must occur somewhere in the system. In the model shown with the reference point on the hull, this motion produces an error in the measurement of displacement. The frequency of this motion, being in the same range as the frequency of interest, makes the desired and disturbance forces indistinguishable. In the model shown with the reference point on the propeller shaft, the disturbance motion does not affect the measured displacement, but may produce the propeller added mass and damping effect previously discussed.

If the bearing disturbance frequency is near the mass-spring natural frequency, motion occurs across the spring. With the reference point on the hull, this motion will not affect the measured displacement if the disturbance motion is absorbed by the spring and does not affect the mass. For the case of the reference point on the shaft, this motion does affect the measured displacement but has a frequency sufficiently above the desired force frequency that it may be discriminated.

From this analysis, the authors conclude that the selection of the analytical model with the thrust measurement reference point on the propeller shaft is preferable.

The proper location of the reference point for torque measurement is dependent upon the type of transducer. A general consideration, however, is the fact that any lateral motion of the shaft may influence the measurement. This motion could be caused by a change of shaft position in the support bearings, eccentricities in the shaft, or dynamic deflection caused by whirling.

The preliminary determination of the allowable actuating device stiffness for thrust and torque measurement is made

from equations (1) and (2) which relate the parameters -- measurement system natural frequency, stiffness, and mass or moments of inertia.

$$\omega_n^2 = \frac{k_1}{m} \qquad \text{(thrust)} \qquad (1)$$

$$\omega_n^2 = \frac{k_2(I_1 + I_2)}{I_1 I_2} \qquad \text{(torque)} \qquad (2)$$

Assuming a value of $m = 2.5$ oz. and $I_1 = 400 \times 10^{-7}$ in. lb. sec.^{2} (includes propeller, propeller hub, entrained water, some shafting, and possibly part of the actuating device), a value of $I_2 = 720 \times 10^{-7}$ in.lb-sec.^{2} (mainly that of a suitable flywheel), and with the value of $\omega_n = 2000$ cps previously selected, the following values are obtained:

$k_1 = 63{,}900$ lb./in. (thrust)
$k_2 = 4060$ in.-lb./radian (torque)
$= 65{,}000$ in.-oz./radian (torque)

The measured displacements are obtained from equations (3) and (4) for the various input forces of interest determined

in the Input Section of this paper.

$$x = \frac{T}{k_1} \tag{3}$$

$$\phi = \frac{Q}{k_2} \tag{4}$$

These values of displacement are (assuming a single spring):

$x = 5.0$ microinches	(thrust = 0.32 lb.)
$= 0.05$ microinches	(thrust = 0.0032 lb.)
$\phi = 20.3 \times 10^{-6}$ degree	(torque = 2.3 in.-oz.)
$= 0.0814 \times 10^{-6}$ degree	(torque = 0.0092 in.-oz.)

These values of displacement are intended to be representative only and are entirely dependent upon the system configuration and analytical models upon which equations (1) and (2) are based. Appendix C presents the detailed considerations for these configurations. In an actual preliminary design of a dynamometer this cycle would be repeated and refined. This refinement, or optimization, procedure would involve various compromises such as sensitivity versus actuating device stiffness, system stiffness versus system mass and moments of inertia, and natural frequency versus harmonic amplitude and phase distortion.

f. Selection of spring types and determination of strains

The use of deflection of a spring element or the strains developed in the spring as a measure of the applied torque and thrust has been established. The deflections were determined by the dynamic analysis in the preceding sections, and transducer configurations are determined in the following sections. In order to determine the strains, however, these deflections must be applied to a particular actuating device.

Desirable spring characteristics for strain determination are: low hysteresis, repeatability, low thermal sensitivity, and linearity. These characteristics are obtained not only by the selection of a suitable configuration, but by the proper selection of spring materials and possibly by providing a controlled environment in which the actuating device operates.

There are many different types of actuating devices available. Seismic mass and spring types have been eliminated by the authors. Bellows, diaphragms, shaft rotations and rectilinear motions, proving rings, and cantilever beams are possible types to produce a measurable strain. The low values of the desired quantities force the selection of

7. Selection of spring types and determination of weights

The use of deflection of a spring element as the sensing element in the spring is a measure of the applied forces and torques has been established. The deflections were determined by the dynamic analysis of the operating machines, and tolerances configurations are determined in the following sections. In order to determine the elastic, however, these definitions must be applied to a particular measuring device.

Desirable spring characteristics for simple determination are: low hysteresis, reproducibility, low thermal sensitivity, and linearity. These characteristics are obtained not only by the selection of a suitable configuration, but by the proper selection of spring materials and quality by providing a controlled environment in which the measuring device operates.

There are many different types of measuring devices available. Simple beam and spring types have been eliminated by the authors. Bellows, diaphragms, which vibrated and cantilever beams, torsion rings, and capillary beads are possible types to produce a measurable signal. The low values of the desired quantities favor the selection of those types producing the greatest strain. In general, cantilever beams provide greater strain amplification than the walls of a thin cylindrical test section. Both types are considered, however, because four previous attempts to design a sensitive dynamometer utilized a thin wall section (8,9,10,11).

Diaphragms are also considered for use as actuating devices in conjunction with several transducer types. References (27) and (28) should be consulted for specific design details and analytical methods.

The following equations relate the various parameters involved in the determination of measured strains:

1. Thrust measurement

$$\frac{x}{T} = \frac{L^3}{3EI} \qquad \text{(cantilever beam)}$$

$$\epsilon = \frac{TLc}{IE} \qquad \text{(cantilever beam)}$$

$$T = \sigma A \qquad \text{(cylinder)}$$

$$\epsilon = \frac{T}{EA} \qquad \text{(cylinder)}$$

2. Torque measurement

$$\emptyset = \frac{GI_p}{l} \qquad \text{(cylinder)}$$

$$\gamma = \frac{QR}{I_p G} = \frac{Q \cdot R\ (2)(1 + \mu)}{I_p E} \qquad \text{(cylinder)}$$

where:

x = axial displacement

T = thrust

L = cantilever beam length

E = Young's modulus

I = moment of inertia of beam cross-section

ϵ = axial strain

c = beam half-thickness

σ = axial stress

A = cylinder cross-sectional area

$\emptyset$ = torsional displacement

G = modulus of rigidity

I_p = cylinder polar mount of inertia

l = torque actuating device length

γ = torsional strain

Q = torque

R = torque actuating device maximum radius

μ = Poisson's ratio

A procedure for the use of these equations to determine actuating device parameters and representative values of strain is shown in Appendix C. Representative values of strain for thrust measurement, using a four cantilever configuration, are:

= 2.21 microinch/inch (thrust = 0.32 lb.)

= 0.0221 microinch/inch (thrust = 0.0032 lb.)

The designer of a specific type actuating devices will find a comprehensive survey of torsion and flexure device theory, design, construction, and use in references (29), (30), and (31).

3. Transducer and Signal Conditioners

At this point of the preliminary design of the sensitive propulsion dynamometer, the problem has been carefully defined and a detailed description of representative measurable quantities has been given. The selection of a transducer and its associated signal conditioners to convert these measurable quantities to a readable signal is usually a compromise among such factors as cost, availability, basic simplicity, reliability, and low maintenance. The

magnitudes of the desired quantities for the small ship model are so small and the frequency requirements so stringent, however, that specific consideration must be given to more than just instrumentation which has been used for measuring quantities of greater magnitude or less stringent frequency requirements. Thus, many of the above factors may have to be sacrificed for even a means of effecting the measurements.

It is evident from the nature of the quantities developed by the actuating device that the use of transducers which convert these quantities to their electrical analogs has important advantages. These advantages are: 1) The mechanical and electrical transducer principles are inherently compatible. 2) The transducers are usually susceptible to miniaturization. 3) Amplification or attenuation may be easily obtained. 4) Mass-inertia effects are minimized. 5) Remote indication or recording is feasible.

At the outset of the search for the suitable transducer, the authors were quite unfamiliar with transducing techniques and the associated signal conditioning. Because

of this unfamiliarity, many techniques were investigated to great lengths in pursuit of sufficient understanding upon which to evaluate their application to the small ship model. Much time, in fact, was spent evaluating techniques which, to the experienced instrumentation designer, would have been obviously unsuitable. In most instances sufficient information on which to base a complete evaluation was not available in the transducer references. Original source articles were consulted but the applications reported, though helpful, were for the measurement of much larger displacements or for less stringent frequency requirements. Extensive investigation of this nature still revealed no conclusive evidence that the desired measurements could be made. It became evident that in order to effect the measurements, techniques developed at or near the state of the art would be necessary. Thus, the authors made inquiries at the M.I.T. Instrumentation Laboratory where it was believed that current experimentation in the field might possibly be in progress and such information could be obtained.

Mechano-electrical transducers may be classed into four main groups: variable parameter analog, self-generating analog, frequency or pulse generating, and digital. Analog transducers are those which produce an electrical output that is a proportional continuous measurement of the input parameter variations. Pulse rate transducers are those which produce voltage pulses whose frequency or random pulse generation rate is proportional to the input parameter variations. Digital transducers are those which produce a unique coded current or voltage form for each discrete value sensed. As the input parameter varies, new values of the coded signal are generated by the transducers (13).

The principles of operation of the transducers investigated within each of the four main groups are briefly described in Appendix D. This Appendix is presented with the intention that it would give the reader a familiarity he may not already possess. The specific transducer references given in the Appendix should be consulted, however, where specific details are desired.

The signal conditioner provides the necessary link between the transducer and data readout equipment and includes all system elements that are used to perform necessary and distinct operations in the measurement sequence. The means of signal conditioning may be classed as input modification and instrumentation amplification. Methods of removal of the signal from the shaft, although not means of conditioning the signal, are a necessary link and will be discussed with signal conditioners. Input modification may accomplish any of the following: conversion of the output into a voltage, current, or digital code; straightforward amplification of the transducer output; filtering out of unwanted frequencies from both transducers and associated circuitry; and impedance matching or signal attenuation. Instrumentation amplification comprises the mostly commonly used signal conditioning circuits and may be amplification of total amplitude, a-c component only, d-c level only, or either Amplitude Modulated or Frequency Modulated carrier type. Modulation is widely used to simplify the amplification of d-c or low frequency signals.

A brief description of signal conditioning devices and applications is given in Appendix E. This Appendix is also presented with the intention that it would give the reader a general familiarity with signal conditioning devices. The references given should be consulted where specific details are desired. The methods of signal removal are discussed in Appendix F.

The transducers which show particular promise for application to the small ship model or which were the specific subject or recommendation of past theses, along with their associated signal conditioners, are analyzed in this section.

a. Strain gages

Strain gages are widely employed to measure the axial and torsional forces in a rotating shaft. Four attempts have been made at M.I.T. to utilize strain gage techniques to measure the torque and thrust developed by a small ship model (8,9,10,11). For these reasons, strain gages have been analyzed to determine their applicability to the sensitive propulsion dynamometer.

Various strain gage actuating device configurations may be used. Several configurations are discussed and

analyzed in Appendix C. This analysis produced the following representative values of strain developed by the desired input thrust:

= 2.2 microinch/inch (for $T = 0.32$ lbs.)
= 0.0221 microinch/inch (for $T = 0.0032$ lbs.)

These values of strain are not to be interpreted as final values. Even if they were final values, the strain gage itself would probably not "see" these values due to such effects as bond stiffening and cross-sensitivity.

Strain gages are generally used most advantageously in a Wheatstone bridge circuit. The maximum possible bridge constant is 4 (four active bridge elements). Typical values of strain gage factors, G.F., are:

G.F. = 2.0 for foil type bonded strain gages
G.F. = 100 for piezoresistive (semiconductor) strain gages

The value of the bridge circuit output voltage per volt excitation is obtained from the analysis procedure of reference (32):

$$\frac{\Delta E}{V_{exc.}} = (G.F.)(\epsilon)$$

$= 0.0044 \text{ mv}/V_{exc.}$ (highest thrust of interest, foil gage)

$= 0.000044 \text{ mv}/V_{exc}.$ (lowest thrust of interest, foil gage)

$= 0.22 \text{ mv}/V_{exc}.$ (highest thrust of interest, semiconductor)

$= 0.0022 \text{ mv}/V_{exc}.$ (lowest thrust of interest, semiconductor)

The resultant signal voltage is dependent upon the excitation voltage applied to the bridge circuit. The limitation on excitation voltage for any strain gage is its heat dissipating capability. The strain gage mounting surface must act as an effective heat sink in order to operate with the higher currents produced by higher applied voltages. Semiconductor strain gages, in addition to having higher sensitivities, can be made with higher resistance values than foil gages which enables them to be excited by higher voltages. Semiconductors, however, are extremely temperature sensitive, that is their sensitivities change

with temperature variations. This effect can be reduced by properly incorporating them into a temperature compensated bridge arrangement, as discussed in most strain gage texts.

In addition to the configurations presented here, Mason and Thurston (33) show the results of the use of a small experimental, single-crystal, n-type germanium torsional transducer which acts as its own actuating device. Their reference to the "negligible angular displacement of the transducer itself", under applied torque, does not define the spring constant of the crystal; but to use an equally vague expression seen frequently in the literature, the crystals, in general, display very high natural frequencies. Mason and Thurston do, however, present the experimentally determined output voltage-torque characteristic which shows an output of approximately 0.08 mv per inch-ounce of torque. They also measured values of torque down to approximately 0.28 inch-ounces. There was no indication given as to the accuracy of these low values, but the values are in the same range of magnitude as the mean torque developed by the small ship model. This

technique merits further investigation into its applicability to the measurement of the small ship model mean values of torque.

Unbonded wire strain gages also find wide application to measurement devices. Reference (34) explains an application in a flush-diaphragm pressure transducer in which the wire acts as its own spring. This, however, requires a certain amount of pre-tensioning of the wires which reduces their sensitivity. The sensitivity reported is in the same range as that given for foil type bonded strain gages.

Reference (27) states that strains of one microinch/inch are detectable with commercial equipment. If this can be interpreted as being the lowest resolution obtainable, it becomes evident that strain gages are unsuitable for measurement of the small forces developed by the small ship model. This reference may, however, have been made to the capabilities of strain gages other than semiconductors. The authors found no other current reference to the lowest values of strain detectable to clarify this point.

As pointed out, strain gages are generally used in a bridge circuit. If sliprings are utilized to transmit the

signal off the rotating shaft, the variations in slipring resistance can be as great as the variations in the strain gages themselves. The current flow through the legs of a bridge is higher than the flow in the output circuit. For this reason, the full bridge must be located on the rotating shaft with the slipring in the external circuit. This is discussed in more detail in Appendix F, along with other methods for removing the signal from the rotating shaft.

b. Linear Variable Differential Transformer (LVDT)

The principles of operation and the techniques for the use of LVDT's are very thoroughly discussed in reference (35). Bortner and Stabile (8) attempted to apply this device to the measurement of thrust in a sensitive propulsion dynamometer. Their work indicated that an LVDT may be suitable for measuring the mean values of thrust. They obtained fairly good sensitivities, however, by using a very soft spring, totally unsuitable for the measurements of the thrust variations. In addition, their spring configuration was unsatisfactory due to the nonlinearities in its output. Possibly a metal diaphragm would have been a more suitable actuating device. Their signal off. On the rotating shaft, the variations in slip-ring resistance can be as great as the variations in the strain gages themselves. The current flow through the legs of a bridge is greater than the flow in the output circuit. For this reason, the full bridge must be located on the rotating shaft with the slip-rings in the output circuit. This is discussed in more detail in Appendix F, along with other methods for removing the signal from the rotating shaft.

5. Linear Variable Differential Transformer (LVDT)

The principles of operation and the techniques for the use of LVDT's are very thoroughly discussed in reference (33). Dayton and Jacobs (1) attempted to apply this device to the measurement of torque in a propulsion dynamometer. Their work indicated that an LVDT may be suitable for measuring the mean values of torque. They obtained fairly good sensitivities, however, by using a very stiff section, totally unsuitable for the measurement of the torque variations. In addition, their spring configuration was unsatisfactory due to the nonlinearities in the output. Possibly a better transducer would have been a more suitable mechanical system. Their general configuration, however, will be the one selected for this discussion.

Reference (35) states that the movable-core differential transformer pickup sensitivity varies with the carrier frequency of the current in the primary coil. For application to this design, the carrier frequency must be at least 2000 cycles per second, or ten times higher than the highest frequency of interest. For this carrier frequency, a Schaevitz Model 050HR LVDT has a sensitivity of 0.003 millivolt per microinch with six volts of excitation (36). This would produce an output signal of 0.015 millivolt for the highest mean thrust of interest, and 0.00015 millivolt for the lowest value of interest. Reference (37) claims a sensitivity of 0.3 millivolts per microinch displacement with 24 volts d-c excitation for a DC LVDT. This would produce an output signal of 1.5 millivolts for the highest mean thrust and 0.015 millivolts for the lowest value of interest.

These values can now be compared to the variation of voltage due to sliprings of the type discussed in Appendix F. A low noise slipring is discussed which has a variation general configuration, however, will be the one selected for this discussion.

Reference [35] states that the variable-core differential transformer (linear) sensitivity varies with the carrier frequency of the current in the primary coil. For application to this design, the carrier frequency must be at least 5000 cycles per second, or ten times higher than the highest frequency of interest. For this carrier frequency, a Schaevitz Model (type) 050 has a sensitivity of 0.003 millivolt per microinch with six volts of excitation [36]. This would produce an output signal of 0.03 millivolt for the highest peak signal of interest, and 0.00002 millivolts for the lowest value of interest. Reference [37] states a sensitivity of 0.3 millivolts per microinch displacement with 20 volts a-c excitation for a 60 type. This would produce an output signal of 3.0 millivolts for the highest peak signal and 0.002 millivolts for the lowest value of interest.

These values can now be compared to the variation of voltage due to slippage of the type discussed in Appendix F. A flat voltage slippage is discussed which has a variation of 3 milliohms resistance. Reference (37) states that excitation input current is 25 milliamps for the d-c transducer. This produces a voltage fluctuation of 0.075 volts which swamps all the above mentioned signal voltages except that put out by the DC LVDT for the highest value of thrust. This problem may possibly be circumvented by the use of some of the techniques presented in Appendix F.

Lion (38) stated, in 1959, that the smallest displacement to be detected with an LVDT was in the order of 10^{-5} inches. Since it is not know if this statement was made after the development of the DC LVDT just discussed, displacements smaller than 10^{-5} inches be detectable.

c. Variable inductance

The most promising variable inductance device for application to the small ship model is that in which the reluctance of a small air gap in a magnetic circuit is changed by displacement of the actuating device. This principle has been used successfully for measurements in large models (6) and with varying degress of success in the small ship model (12).

The "magni-thrust" and "magni-torque" elements, described in reference (6), are a basic design for DTMB transmission dynamometers. These elements, like the magnetic-coupled torquemeter described in reference (39), do not require sliprings. The coils in which the varying inductance is detected completely encircle the shaft and the magnetic return paths are through encasing shells of magnetic material and across the radial air gaps between the stationary coils and the rotating shaft. Since the radial air gaps have a full circumferential area, which is large compared to the area of the active air gaps, the variations in the radial air gaps are negligible compared to the variations in the active air gaps on the shaft.

Ricketts and Flaherty (12) designed, constructed, and tested a variable reluctance device for measuring thrust on the small ship model. The device, requiring slip rings, consisted of two coils wound inside Ferroxcube pot cores which were spring mounted on the rotating shaft in such a manner that the thrust caused an increase

in the air gap of one pot core and a decrease in the air gap of the other. Each of the varying inductances was placed in a separate resonant circuit whose frequency was dependent on the values of the inductance. The difference in the frequencies of the two resonant circuits was an indication of the thrust developed. Test results of this device showed a high overall sensitivity of 908 millivolts per pound of thrust.

The authors are unable to place much confidence in the high value of sensitivity of the Ricketts and Flaherty device since neither proof of a correct spring constant was given nor was the sensitivity obtained through dynamic calibration. The Ricketts and Flaherty developments for application of the principle to the small ship model for measurement of thrust and a proposed configuration for measurement of torque are excellent and should be consulted for transducer and signal conditioning techniques in application of the variable reluctance principle. A configuration was also proposed which incorporated the Ferroxcube cores without slip rings; however, the design was such that the fixed

reference point was off the shaft. For reasons previously discussed, such a configuration is unacceptable.

In the search for further evidence on which to evaluate the use of a variable reluctance device, several sensitivity figures were found. According to reference (40), a typical E-magnet angular displacement receiver having a volume of two cubic inches and a weight of six ounces and excited with a 60 cycle per second, 50 milliampere current, operates over a range of input angles of 0.1 radian with an open-circuit angle-voltage sensitivity of 300 millivolts per milliradian. Referred to the angular twist developed by the mean torque and the smallest torque variation of interest,

$$\begin{aligned} E_o(\text{open circuit voltage}) &= 0.10 \text{ mv } (Q = 2.3 \text{ in.-oz.}) \\ &= 0.00042 \text{ mv } (Q = 0.0092 \text{ in.-oz.}) \end{aligned}$$

Axial and radial play in bearings are presently being measured at the M.I.T. Instrumentation Laboratory using a "magnetic suspension" device with a sensitivity of 2.2 millivolts per microinch. Referring this sensitivity to the torque and thrust values of interest:

E_o = 11.0 mv (T = 0.32 lb.)

= 0.11 mv (T = 0.0032 lb.)

= 17.5 mv (Q = 2.3 in.-oz., r = 1/2 in.)

= 0.07 mv (Q = 0.0092 in.-oz., r = 1/2 in.)

In the work at the Instrumentation Laboratory, resolution of displacements of approximately one-tenth of a microinch is expected. The signal conditioners used for these measurements are not known, but the fact that the measurements are being made in this range, although in the Laboratory, is encouragement that at least the mean values of thrust and torque might be measured.

d. "Geared torque dynamometer" and similar configurations for torque and shaft rotational velocity measurements.

Ricketts and Flaherty (12) also presented a "geared torque dynamometer" configuration for measuring flexure twist angle. The twist angle was to be measured as the phase difference between the digital outputs of two magnetic pickups actuated by multiple toothed gears placed on opposite sides of the flexure. The authors believe, and the following

development shows, that the expectations of the "geared torque dynamometer" were overly optimistic.

Using the circuitry proposed for the configuration, the frequency of the output of the magnetic pickups represents the frequency at which the torque variations would be sampled. Practical sampling theory requires that this frequency be at least ten times the highest frequency of interest. Since the sampling frequency also equals the number of gear teeth times the shaft rotational velocity, the number of gear teeth required is calculated as follows:

Sampling frequency = 10 x highest frequency of interest
= shaft rotational velocity x
number of gear teeth

Substituting values from Table II and solving,

Number of gear teeth required = 10 x 200 cps/20 rps
= 100 teeth

A procedure for determining the optimum tooth size and spacing for the maximum output of a magnetic pickup is given in reference (41). Using this procedure and the smallest pole piece commercially available, the following

gear and pickup dimensions were calculated:

Maximum gear diameter	2 in.	(model restriction)
Width of gear teeth	0.0157 in.	
Gap between teeth	0.0471 in.	
Height of teeth	0.0471 in.	
Thickness of gear disk	0.040 in.	

The use of this gear further requires that the pole piece, 0.040 inches in diameter, be machined to a chisel point of 0.0157 inch thickness. The dimensions of the gear and the pole piece are obtainable, but the manufacturing tolerances are critical. The teeth must be spaced exactly uniformly on the circumference of the gear disk and the two gears must be identical. A variation of only 0.14 microinch in the width between two adjacent teeth will produce a torque variation equal in magnitude to the smallest torque variation of interest. For accurate measurements of the torque variations, such allowable tolerances in the machining of the gear teeth would produce 100 o/o error. So-called precision digital gears, currently being used in work at the M.I.T. Instrumentation Laboratory, have 415 teeth on a disk of

1.688 inch diameter. These gears are generated by Arch Gear Works, Quincy, Massachusetts. A representative of that company stated that the composite error between teeth can be controlled only within 0.0002 inch and the roundness of the gear to a tolerance of 50 microinches. Thus, tolerance considerations alone discount using the "geared torque dynamometer" for effecting measurement of the desired quantities.

The photoemissive cell or phototube could also be used in a digital configuration similar to that of the "geared torque dynamometer". The phototubes would receive light energy reflected from mirrored surfaces on both sides of the shaft flexure element. The number of surfaces and the tolerance of these surfaces, however, would require precision machining equal to that required for the gears.

Another similar configuration might use the photoconductive cell to measure variations in light intensity caused by angular twist of the torque flexure. The light intensity would be varied by the angular twist between two shaft-mounted disks in which either radial or spiral

grates were cut. This method may show promise for application to the small ship model and should be investigated further.

Probably the simplest method of measuring shaft rotational velocity is to count the pulses generated as an electromagnetic gear passes a magnetic pickup. The output of the pickup can be fed directly to an electronic counter, or through a frequency to voltage device to a recorder. The output of a 15 tooth gear can be multiplied by four to get revolutions per minute. A similar velocity pulse arrangement could be devised using photoelectric devices. The accuracy of the velocity measurement can be improved by increasing the number of teeth on the gear.

e. Piezoelectric transducers

Piezoelectric crystals have sensitivities up to about 5 millivolts per microinch. A stress resulting in a strain of 10^{-2} microinch per inch may produce as much as 0.2 volts (22). These high sensitivities make them worthy of investigation for their applicability to this dynamometer.

Piezoelectric elements usually act as part of the actuating device. Their fundamental natural frequency

may be as high as 100,000 cycles.per second, but the natural frequency of most commercial piezoelectric transducers is between 25,000 and 75,000 cycles per second. Damping is quite low, about 0.002 to 0.22 of critical (22).

Output impedances are usually in the range of 100 to 10,000 micromicrofarads of capacity. This capacity and the input resistance of the signal conditioner to which the device is attached limit the low frequency response of the transducer. This high output impedance requires that the output be fed directly into a device such as a cathode follower, or other high impedance device, for the measurements of vibrations in the range of those developed by the small ship model.

Single piezoelectric crystals can be cut with the appropriate orientation and electrode application to enable them to be used for either flexing or torsional movements. However, in any propulsion dynamometer configuration, the crystals would be required to transmit both torque and thrust. The directional sensitivity of these devices would have to be determined with extreme care, since

even small amounts of cross-sensitivity would result in significant errors.

Piezoelectric properties vary according to whether the applied load compresses, bends, or twists the elements. Lion (38) discusses various configurations and indicates that the elements have a higher sensitivity, at the expense of mechanical resonance frequency, when used as bimorphs (trade name of Brush Electronics Company)., Bimorphs are termed either benders or twisters, depending upon their application.

Using the methods outlined in Harris and Crede (22), the authors calculated that the spring constant of a representative barium titanate crystal subjected to compression loading would be approximately 9.9 x 10^6 pounds per inch. Draper (40) states a representative value of sensitivity of 5 millivolts per microinch for a compression loaded piezoelectric transducer. If a thrust measuring configuration were to use a transducer with these characteristics, the highest thrust of interest would produce a displacement of approximately 0.032 microinches and a signal voltage of approximately 0.15 millivolts. Since

this actuating device stiffness is 150 times higher than required by the dynamic analysis discussed earlier, a bimorph bender configuration with its decreased stiffness and higher sensitivity would seem more appropriate than a compression device. However, Draper states that a typical bender sensitivity is 0.25 millivolts per microinch displacement. This appears to contradict the arguement presented in favor of benders. The apparent contradiction may be due to the use of different materials, element sizes, or signal conditioners.

The authors' analysis of piezoelectric transducers is not sufficient to properly evaluate their application to the small ship model sensitive dynamometer. Investigations into current laboratory techniques in the use of these transducers may reveal additional information on which such an evaluation could be made.

f. Variable capacitance transducer

The output of a displacement-sensitive capacitance transducer is proportional to the change in capacitance caused by the relative displacement between a conducting plate and the actuating device. Appropriate signal

conditioners are used to generate a voltage corresponding to this change in capacitance. Since change in capacitance with respect to plate separation is the most sensitive, only that type is considered for application to measurements in the small ship model.

Harris and Crede (22) state that the main advantages of the capacitance transducer are: 1) simplicity in installation, 2) negligible effect on the operation of the vibrating system, 3) extreme sensitivity, 4) wide displacement range, and 5) wide frequency range. The disadvantages of the transducer are: 1) inherent complexity of associated signal conditioners, 2) relatively large output impedance which requires careful shielding and short connecting cables, and 3) nonlinearity in the relationship of the output voltage to mechanical displacement. In spite of these disadvantages, however, the advantage of extreme sensitivity alone places this type of transducer as possibly the most promising type for effecting the desired measurements. Lion (38) states that these transducers have been used for the measurement

of displacements as small as about one-tenth of an angstrom unit (4×10^{-11} inch).

Before the transducer can be applied to the small ship model, correction of the disadvantages must be considered. The principal sources of the nonlinearity (22) are: 1) the effect of fringing in the electric field between the capacitor plates, 2) the fact that the capacitance is inversely, and not directly, proportional to the spacing, and 3) the electric circuit associated with the transducer.

The effect of the fringing depends to such an extent upon the transducer plate design and its shielding that it is difficult to determine analytically. Fringing effects should be checked experimentally for each installation.

The capacitance variation, which is obtained when the spacing between the plates is changed, is not a linear function of the spacing. This can lead to serious distortion in the output in cases where the minimum displacement which must be resolved is less than about one per cent of the maximum displacement. For the application to the small ship model, the minimum displacement which must be

resolved is the maximum tolerable error shown in Figure 1. That value for accuracy within ± 0.25 per cent is 23.0×10^{-6} in.-oz. for torque and 5.0×10^{-6} lb. for thrust. These values are much less than one per cent of their respective mean values, 0.97 in.-oz. and 0.135 lb. given in Table II. In order to obtain a maximum tolerable error value that is one per cent of the mean values, the accuracy must be relaxed such that error is greater than the variation to be measured. As shown in reference (42), however, the linearity of the capacitor can be improved by inserting mica, a material with a high dielectric constant, such that it partially fills the gap. The correct thickness to obtain linearity is found by methodical experimentation. The addition of the mica improves not only the linearity, but also the frequency response, by adding damping, and the robustness of the device, by adding insulation.

The magnitude of the output impedance depends on the frequency of the alternating current used for the determination of the capacitance; for practical cases realized in the maximum harmonic error shown in Section 1. Thus using the frequency stability ± 0.03 per cent is ±1.9 x 10^{-4} in-oz. for torque and 1.0 x 10^{-4} lb. for thrust. These values are well less than one per cent of their respective full scales, 0.59 in-oz. and 0.133 lb. given in Table II. In order to obtain a maximum harmonic error value that is one per cent of the above values, the capacitor must be selected such that error is greater than the variation to be considered. As shown in reference 1, however, the linearity of the capacitor can be improved by inserting also, a material with a high dielectric constant, such that it partially fills the gap. The current technique to obtain linearity is based on capacitor configuration. The thickness of the disc improves not only the linearity, but also the frequency response, by adding damping, and the ruggedness of the device, by reduced lamination.

The magnitude of the output impedance depends on the frequency of the alternating current used for the determination of the capacitance; for practical cases (capacitances of the order of 10 to several hundred μμF), the output impedance is in the range between 10^3 and 10^7 ohms. The magnitude of both the output signal and the output impedance may be changed by series and parallel capacitor circuits, but such modifications are always accompanied by a reduction of the signal (38).

The dynamic response characteristic of capacitive transducers, as well as hysteresis, mechanical after effects, and drift, and the influence of the environmental temperature and pressure upon the transducer performance are all determined by its mechanical construction rather than by its electrical characteristic. The source of the greatest mechanical difficulties is frequently the insulation employed to hold the plates in position. Capacitive transducers have been used for a wide variety of measurements of physical magnitudes and details of the constructions used are described in references listed in the capacitance bibliography of reference (22). Most of these references were read by the authors and found to be excellent application examples.

The minimum resolvable displacement is virtually unlimited, depending only on the maximum pickoff area which can be used and the minimum air gap which can be constructed. Roberts (42) states it is not feasible in practice, however, to use smaller gaps than 0.001 inch. He also discusses a push-pull configuration employing three plates, one moving and two fixed, by which measurement is made in terms of the ratio of the two capacitances, one increasing and the other decreasing. So long as the ratio between them is unchanged, changes in the value of the capacitance from thermal expansion of the parts or from changes in the dielectric are of little consequence.

In general, greatest overall stability is possible using the balanced bridge circuit to convert the change of capacitance to a voltage. The size of the capacitive elements, however, complicates placing and balancing the elements on the rotating shaft. It may appear that only the active element need be placed on the shaft since variations in the resistive component of the impedance, contributed

by the sliprings, would have negligible effect on the value of the impedance, which is controlled by the capacitance reactive component. Capacitance in the sliprings, possibly in the order of picofarads, may be of the same order as the variations of the capacitive element. The bridge, therefore, must be placed on the shaft and one of the methods of signal removal from the shaft must be employed.

A thrust configuration employing a capacitive transducer and a resonant circuit on the shaft is shown in Figure 5. The method of signal removal from the shaft is by a rotary transformer arrangement, thereby eliminating the need for sliprings. The circuit is so designed that a variation in the capacitor air gap causes a capacity which in turn modifies the resonant frequency of the tuned circuit. When this frequency changes with respect to the fixed-frequency oscillation of the excitation supply, the associated electrical circuits produce a voltage change that is picked up by the inductively coupled winding and then rectified. This rectified voltage is the output signal of the thrust transducer. The effects of variations

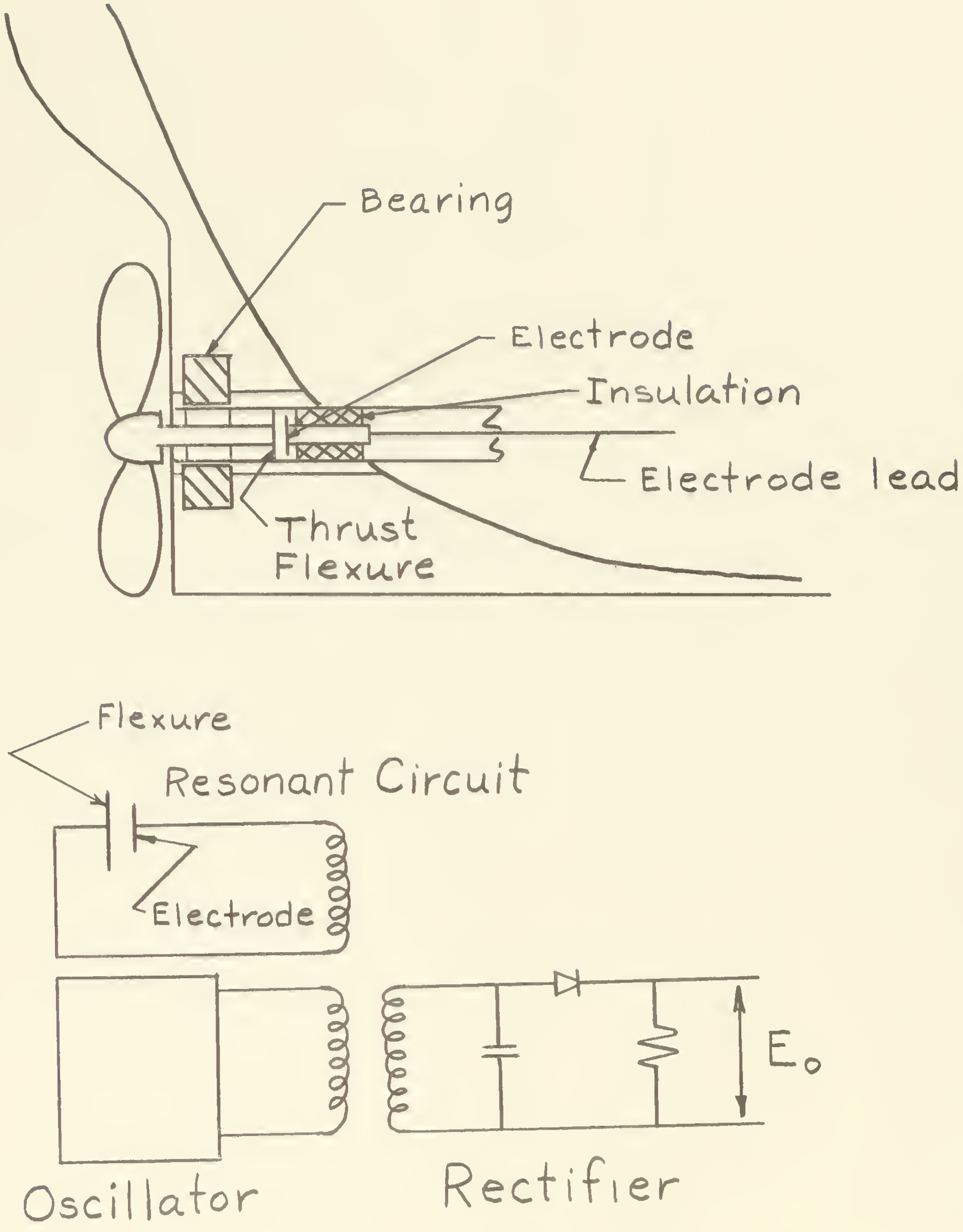


Figure 5. Possible Variable Capacitance Transducer and Signal Conditioner Configuration for Thrust Measurement

in the gap of the inductive coupled slipring remain for future investigation.

Torque configurations using similar principles can be devised. One such configuration might employ capacitance transducers on the circumference of a torque flexure. Thus, the angular twist would vary the plate separation in the same manner as the push-pull arrangement described. A serrated-capacitor type transducer, described in reference (43), might also be modified to apply to the small ship model.

g. Electron tube transducer

The electron tube transducer is a variable resistance device in which a voltage change is developed when the relative spacing between the movable plate and the fixed grid is changed. The change is actuated through suitable linkage between the actuating device and the shaft supporting the plate of the tube. The transducer has several principal disadvantages for application to the small ship model: 1) unpredictable "zero drift", 2) "zero shift" following impacts of short duration, and 3) difficulty in hermetically sealing such that controlled vibrations can

be transferred. In spite of these disadvantages, however, the tube is considered for its high sensitivity, the most important requirement to effect the desired measurements.

The RCA Type 5734 mechanoelectronic transducer, described in the transducer references of Appendix D, occupies 0.5 cubic inch, weighs in the region of 0.25 ounce, and can be mounted in any position. Placed in the voltage-sensitive circuit described in Appendix E, with a load resistor of about 50,000 ohms and a plate supply of 20 volts, the displacement-voltage sensitivity of the system is in the region of 5 millivolts per microinch over an input range of 0.001 inch. This sensitivity gives an output of 0.25 millivolt for the thrust variations. In the nonactuated position, the resistance of the tube is approximately 75,000 ohms and the change in resistance is 125 milliohms per microinch. The magnitude of the resistance, 75,000 ohms, is rather large for use in a balanced bridge circuit installed on the shaft; therefore, a means of signal removal from the shaft must be used in the voltage-sensitive circuit. The following calculations for the circuit using sliprings are presented for comparison:

Change in slipring resistance = 3 mΩ (Appendix F)

Change in tube resistance
for thrust variations = 125 mΩ/μin. x 0.25 μin.
= 6.25 mΩ (T = 0.0032 lb.)

Thus, the change in slipring resistance is approximately one-half the change in tube resistance due to the thrust variations. Whether or not the slipring noise could be filtered out of the signal depends on the frequency of the noise compared to the frequency of interest.

Frequent calibration, at the least, may be necessary in order to overcome the disadvantage of unpredictable "zero drift". If this disadvantage can be overcome, and sufficient resolution obtained through the signal conditioners, then the transducer might be applied to thrust measurements in the small ship model. The fact that the units are fragile and require special handling also present a problem requiring further investigation. To circumvent the slipring problem, either commercially manufactured mercury sliprings or the wire wound technique of signal removal from the shaft, discussed in Appendix F, might be used.

h. Selsyn Systems

Selsyn systems, described in reference (44), were investigated extensively for use as the torque transducer. In general, it was found that accuracy of these devices was inadequate for the measurements desired. The best accuracy available in a two-speed synchro-system employing the best possible gearing is about ± 0.0083 degree. References (45,46,47) give excellent treatments of the design and application of Selsyn systems.

Investigations at the M.I.T. Instrumentation Laboratory, however, revealed excellent results with application of the Microsyn. Microsyns are quite temperature sensitive when attempts for accurate measurements are made. Under ideal temperature control and static conditions, angles of 2.78×10^{-6} degrees can be distinguished. Where temperature is controlled to within 0.01 degree Fahrenheit, angles of 1.39×10^{-6} degrees can be distinguished. For comparison, the angular twist for the highest mean torque of interest ($Q = 2.30$ in.-oz.) is 20.3×10^{-6} degrees and for the smallest torque variation of interest ($Q = 0.0092$ in.-oz.) is 0.08×10^{-6} degrees. Thus, if ideal conditions could

be met, Microsyns could only be used for mean torque meansurements with approximately 14 o/o maximum error.

Comparative information for the other specialized types was not available. Further investigation may reveal that the others, particularly Inductosyns, are more accurate. It is doubtful, however, that the accuracy of these, under ideal conditions, will be sufficient to effect the desired measurements.

4. Readout devices

The termination of the signal conditioners is a readout device which receives the output signal and presents it in a readout form. This readout may be in two possible forms: relative displacement or digital. Examples of a relative displacement readout device are a meter pointer moving over a scale and a pen or light beam writing a permanent record on a moving paper. Examples of a digital readout are electronic decade counters, and rotating drum mechanical counters.

The nature of the signal output of a small ship model sensitive dynamometer requires some form of a voltage-

sensitive readout device. Various possible electrical indicators and recorders, and their applicability to this measurement system, are briefly discussed. Readout device capabilities are those given in Beckwith and Buck (27), unless otherwise indicated.

a. Meter indicators

Meter indicators have a relatively high meter movement inertia. In addition, it would be impossible for the human eye to follow the pointer even if it were possible for the meter to follow the dynamic signal. For these reasons meter indicators are unsuited for the measurements of the input force fluctuations, but may have application to the measurement of the steady, or slowly fluctuating, mean value of torque and thrust.

b. Mechanical counters

Mechanical counters could be directly connected to the rotating shaft or could be electrically actuated counters energized by switch or relay, or by any source of pulse able to supply a power of several watts. One high speed counter of this type is capable of making 1000 counts per minute. This type of device would produce less

than one count for the shaft rotational velocities of this dynamometer and would probably not be suitable.

c. Electronic counters

Electronic counters are extremely versatile devices, capable of many high speed counting type operations. They have a high impedance input and require little energy for pulsing (0.2 volts rms pulse). This device can be used to accurately measure shaft rotational velocities.

d. Cathode-ray oscilloscope

Cathode ray oscilloscopes are high impedance voltage-sensitive devices with an inertialess beam of electrons substituted for the meter point and a fluorescent screen replacing the meter scale. Electrostatic cathode-ray tube sensitivity is relatively low (about 0.004 to 0.06 inch deflection per d-c volt) and therefore requires an amplified signal. All general purpose oscilloscopes provide such amplification.

If direct observation of the scope does not provide sufficient information about the signal, high-speed photographic records may be made and analyzed. This technique

is described as used at the Netherlands Ship Model Basin, Wageningen, in reference (48).

e. Oscillograph

An oscillograph is a low impedance writing instrument that records the input directly onto paper. There are two basic types available: 1) A direct writing type with a stylus that directly contacts a moving paper strip. 2) A mirror type which employs a light beam for writing on photographic paper or film.

The galvanometer suspension system in an oscillograph limits the frequency response as in any mass-spring system. Table III summarizes typical oscillograph galvanometer sensitivity and frequency values. Table III indicates that a light-beam type galvanometer is the only suitable oscillograph for recording the force fluctuations developed in the propulsion dynamometer. This means that the Sanborn oscillograph with a Model 150 carrier preamplifier currently installed in the M.I.T. Towing Tank is suitable only for the recording of mean torque and thrust values.

Table III

Typical Oscillograph Galvanometer Sensitivity and Frequency Values

Galvanometer	Maximum usable frequency, cps	Sensitivity in./ma.
Stylus-type with amplifying system	100	0.04
Low-sensitivity light-beam type, fluid damped	350	2.3
High-frequency light-beam type, fluid damped	5000	21.7
High-sensitivity light-beam type, magnetically damped	6	61.7

f. Magnetic tape recorders

The capabilities and use of magnetic tape recorders are thoroughly discussed by Harris and Crede (22). These recorders find wide application in the field of vibration measurement and analysis. They provide a permanent record of readout which can be analyzed by many different methods. Several methods are given in the Output section of this thesis.

5. Discussion and other considerations

a. Limitations of analysis

The foregoing analysis has been based on idealized models in which idealized conditions have been assumed. The analysis of the actuating device, for instance, was developed considering that the masses and the frame had infinite stiffness, that the spring had zero mass, and that only viscous damping was present. The transducer and signal conditioning analysis was developed considering the inherent difficulties of the transducer technique itself and the restrictions imposed by sliprings; however, the remainder of the signal conditioning devices were considered within themselves to be ideal. Detailed analyses of mechanical and electrical noises within the transducer and the signal conditioners must be considered before the complete system can be evaluated. It must be considered, therefore, that the performance of an actual system will differ from that predicted for the idealized model.

b. Implications of analysis

It has been pointed out in the analysis that, in order to effect measurement of the desired alternating forces, techniques at or near the state of the art must be employed.

The results of the analyses, however, are insufficient upon which to base complete evaluation. Whereas some of the techniques can definitely be eliminated by the evidence presented, others can neither be eliminated or substantiated conclusively as suitable means.

Four particular techniques standout as most promising, either from presentation of encouraging evidence or from the lack of sufficient discouraging evidence:

1) The photoelectric technique employing photoconductive cells to measure variations in light intensity may in fact be a suitable means for measuring the torque variations on the rotating shaft of the small ship model. A complete investigation of this type, however, remains to be made.

2) The electron-tube transducer has a high sensitivity. If the problems of placing this tube on the rotating shaft with a suitable mechanical linkage and signal transmission arrangement, along with the inherent problems of the tube itself, can be overcome, this device may prove satisfactory.

3) Although information concerning the state-of-the-art techniques of piezoelectric devices is lacking,

recommendations from several sources favor incorporation of this device on the basis of extreme sensitivity. Evidence of extreme sensitivity for small displacements was found, but because of the high spring constant of the device, larger forces are required to actuate the displacement. It is recommended, therefore, that the bimorph type, which gives greater sensitivity at lower natural frequencies, be further investigated.

4) The most promising technique appears to be that of the variable capacitance transducer. The authors believe that a detailed design for construction of a dynamometer using this technique would be beneficial.

Throughout any future work in the design and development of the sensitive propulsion dynamometer it must be borne in mind that the nature of the quantities to be measured require that all phases of design must employ state-of-the-art or near state-of-the-art techniques. This requirement implies that even the simplest component of the system must be constructed or selected with the utmost care in order to produce the highest possible signal-to-noise ratio.

All of the remaining techniques may be capable of measuring the mean values of thrust and torque since for such measurements the frequency requirement could be relaxed and sensitivity would be increased. A reanalysis of the measurable quantities obtainable and the application of the transducer techniques under the relaxed conditions should be made.

c. Compromises

In view of the apparent difficulty in obtaining the measurements of the desired alternating quantities, it is only natural that compromises should be considered. Three types of compromises could be made in an effort to obtain the goal: 1) damping and reduction of the measurement system natural frequency, 2) relaxation of the accuracy requirements, and 3) reduction in the flexibility of dynamometer usage.

Damping and a reduction of the measurement system natural frequency, with subsequent increase in sensitivity, does not involve a compromise in accuracy only if the degree of damping achieved is approximately 65 per cent of critical.

As pointed out in the analysis of actuating devices, this value of damping allows operation to approximately 0.4 of the system natural frequency without harmonic amplitude distortion and produces a phase shift, but no phase distortion. Two particular difficulties are present in the attempt to achieve this amount of damping. First, it is extremely difficult to determine the actual value of the damping obtained; thus, calibration at many different frequencies would be necessary. Secondly, since the frequencies of interest are comparatively low, the large size of the damping devices required could be unacceptable for small ship model application.

Reduction of accuracy requirements might be accomplished in two ways. First, the value of maximum tolerable error could be increased. Figure 1 shows the variation of maximum tolerable error, thus the minimum magnitudes which must be resolved, as a function of the accuracy desired for the measurement of the alternating forces. A compromise of this nature, however, would not help significantly since, in most cases, the transducer techniques analyzed were incapable of

resolving the amplitudes of variation, let alone the value of maximum tolerable error. Secondly, the natural frequency of the measurement system could be reduced with no regard to damping; however, magnitude distortion would result at the various harmonic frequencies of interest. Such distortion would be very difficult to calibrate since it would vary with each different frequency.

Flexibility of dynamometer usage may be reduced by decreasing the highest frequency of interest to be measured, by limiting the speed range of interest for model testing to the higher values, or by limiting the measurements desired to only the mean values of torque and thrust. A decrease in the highest frequency of interest may be accomplished by restricting the number of blades on the propeller to be tested or by arbitrarily eliminating the second harmonic as a harmonic of interest.

Although conclusive evidence has not been given to the effect that the alternating forces cannot be measured, the authors believe that the only practical compromise to be made toward furthering the goal of a propulsion dynamometer

capability at the M.I.T. Towing Tank, is to relax the capability to the measurement of the mean values of torque and thrust only. It is further recommended that such a dynamometer be initially tested in a propeller boat configuration to reduce the installation complications.

d. Other considerations

Other considerations, which have been somewhat neglected in the design analysis of the sensitive propulsion dynamometer, are phase shift and calibration requirements.

Phase shift is the time delay between the mechanical input and electrical output signal of the measurement system. This has been discussed in regard to the distortion which may result if certain requirements are not met, but a method for determination of the shift has not been discussed. A detailed analysis of the mechanical and electrical phase shifts must be accomplished in order to determine their values.

Calibration of the dynamometer involves determining the relationship between the output and the input

of the measurement system. In general, the following information is required:

1) The sensitivity over the frequency range of interest.

2) The sensitivity over the environmental range of interest. For example, the determination of the effects of temperature change, supply voltage variations, stray fields, and humidity.

3) The sensitivity over an amplitude range of interest.

4) The stability of calibration with time.

D. Output

The output of the measurement system is discussed briefly for completeness of the analytical model used and should be further analyzed when an acceptable output can be obtained. The output is dependent on the readout device used and will either be in the form of visual presentation or a time history. Assuming that the self-propelled ship model produces useful output for only the last fifty feet of its tank run and depending on the speed of

the model, the data collection period will be about ten to fifteen seconds.

Data analysis techniques are thoroughly discussed in references such as the Shock and Vibration Handbook, Volume 2, by Harris and Crede (49) and Statistical Theory of Communications, by Y. W. Lee (50). Reference (48) discusses a data analysis technique in use at the Netherlands Ship Model Basin that produces a noise-free composition of the dynamic phenomena observed. Reference (4) discusses the computer aided data analysis techniques used at the David Taylor Model Basin. These references should prove useful for anyone contemplating the design of a sensitive propulsion dynamometer.

IV. CONCLUSIONS

1. It is desirable that the sensitive propulsion dynamometer be capable of measuring the following quantities:

a. Highest frequency of interest = 200 cycles per second.

b. Highest mean values of interest (full speed)

Torque = 2.30 inch-ounces

Thrust = 0.32 pounds

c. Smallest variation amplitudes of interest (65% full speed)

Torque = 0.0092 inch-ounce

Thrust = 0.0032 pounds

d. Highest shaft rotational velocity of interest = 1200 revolutions per minute.

2. Based on the analysis of disturbances or noises:

a. Measurement system natural frequency must be high to eliminate the effect of propeller added mass and damping.

b. Isolation and damping techniques are required to eliminate the effects of prime mover vibrations.

c. A stern tube bearing and shafting system can be designed such that the stern tube torque losses are constant.

3. Based on the analysis of the measurement system:

a. Use of a transmission dynamometer is preferable.

b. A fixed reference instrument must be used with the reference point on the shaft.

c. Strain or displacement are desirable as measurable quantities.

d. The measurement system natural frequency must be at least ten times the highest frequency of interest to prevent amplitude and phase distortion.

e. Techniques at or near the state of the art must be employed to effect measurement of the desired alternating forces.

f. Four particular transducer techniques standout as the most promising for effecting the measurement of the alternating forces:

1) A photoelectric technique employing photoconductive cells to measure light intensity variations resulting from torque variations.

2) An electron tube transducer extremely sensitive to displacements resulting from thrust variations.

3) A piezoelectric transducer extremely sensitive to strains resulting from torque and thrust variations.

4) A variable capacitance transducer representing possibly the best technique for measuring torque and thrust variations.

V. RECOMMENDATIONS

On the basis of practicality, the authors recommend that a sensitive propulsion dynamometer capable of measuring only the mean values of thrust and torque be developed for the small ship model. In order to effect this development, a reanalysis of the elements of the measurement system must be conducted such that the optimum measurable quantity is actuated, that the appropriate transducer and associated equipment is selected, and that the best output analysis techniques are employed.

It is further recommended that the measurement of mean values should be successful before consideration be given to the development of a sensitive propulsion dynamometer capable of measuring the alternating forces. In the event the measurement of the alternating forces is undertaken, the following recommendations are made:

1. Use the Stern Tube Bearing Design Procedure to determine the bearing and shaft design.
2. Conduct a frequency analysis of bearing noises.
3. Further investigate and select the best of the four most promising transducer techniques.

If either type of sensitive dynamometer is to be developed, it is recommended that proper calibration techniques be developed and that the methods of signal removal from the rotating shaft, particularly the direct connection method, be further investigated.

APPENDIX

Appendix A

Determination of Desired Inputs

1. Mean or average values

Parent data of several ships can be scaled to determine anticipated values of mean forces and velocities for small ship models. Model scaling rules used are those presented in reference (51) and are:

$$T_M = \frac{T_P}{\lambda^3}$$

$$Q_M = \frac{Q_P}{\lambda^4}$$

$$V_M = \frac{V_P}{\sqrt{\lambda}}$$

$$N_M = \sqrt{\lambda}\ N_P$$

where T = thrust, Q = torque, V = velocity, N = RPM, λ = scale factor, and subscripts refer to either model or parent.

a. Parent data

Mariner (from reference (52))

LOA = 563.64 feet

LBP = 528.5 feet

V(kts.)	SHP	RPM
14	4000	63
16	6100	73
18	9300	83
20	14,000	94
21	18,500	100

$$Q_p = \frac{550\ SHP}{2\pi N} = \frac{550 \times SHP \times 60}{2\pi N} = 5250\ \frac{SHP}{N}$$

$= 334{,}000$ ft.-lb. (14 kts.)

$= 438{,}000$ ft.-lb. (16 kts.)

$= 589{,}000$ ft.-lb. (18 kts.)

$= 782{,}000$ ft.-lb. (20 kts.)

$= 970{,}000$ ft.-lb. (21 kts.)

Series 60 (from referance (53))

$LWL_1 = 600$ feet (parent 1)

V(kts.)	SHP	RPM
10	2093	40.5
12	3480	48.6
14	5717	57.1
18	13,300	75.8
20	20,900	87.1
22	37,940	103.0

$Q_{P_1} = \frac{550 \text{ SHP}}{2\pi n} = 5250 \frac{\text{SHP}}{N}$

$= 271,000$ (10 kts)

$= 375,000$ (12 kts)

$= 526,000$ (14 kts)

$= 920,000$ (18 kts)

$= 1,260,000$ (20 kts)

$= 1,890,000$ (22 kts)

$LWL_2 = 20$ feet (parent 2)

V(kts)	T(lb.)
10	3.0
16	5.3
22	13.0

b. Model data

Mariner

LOA = 5.895 feet

$\lambda = \frac{563.64}{5.895} = 95.5$

$\sqrt{\lambda} = 9.98$

$Q_M = \frac{Q_P}{\lambda^4} \qquad \lambda^4 = (95.5)^4 = 0.831 \times 10^8$

$$Q_M = \frac{Q_p(\text{ft-lb.}) \times 12 \times 16}{0.831 \times 10^3} \quad (\text{in.- oz.})$$

$= 0.773$ in.- oz. (14 kts.)

$= 1.805$ in.- oz. (20 kts.)

$= 2.24$ in.- oz. (21 kts.)

$$N_M = \sqrt{\lambda}\ N_p$$

$N_M = 628$ RPM $= 10.5$ RPS (14 kts.)

$= 728$ RPM $= 12.15$ RPS (16 kts.)

$= 828$ RPM $= 13.8$ RPS (18 kts.)

$= 938$ RPM $= 15.65$ RPS (20 kts.)

$= 998$ RPM $= 16.6$ RPS (21 kts.)

$$V_M = \frac{V_s}{\sqrt{\lambda}}$$

$V_M = \frac{14}{9.98} = 1.4$ kts.

$= \frac{21}{9.98} = 2.1$ kts.

Series 60

LWL = 5.0 ft.

$$\lambda_1 = \frac{600}{5} = 120$$

$$\sqrt{\lambda_1} = 10.95$$

$$Q_M = \frac{Q_P}{\lambda_1^{4}} \qquad \lambda_1^{4} = (120)^4 = 2.075 \times 10^8$$

$$Q_M = \frac{Q_P(\text{ft.-lb}) \times 192}{2.075 \times 10^8} \quad (\text{in.-oz})$$

= 0.251 in.-oz. (10 kts.)

= 0.347 in.-oz. (12 kts.)

= 0.852 in.-oz. (18 kts.)

= 1.165 in.-oz. (20 kts.)

= 1.75 in.-oz. (22 kts.)

$$T_M = \frac{T_P}{\lambda_2^{3}} \qquad \lambda_2 = \frac{20}{5} = 4 \quad \lambda_2^{3} = 64$$

T_M = 0.0469 lb. (10 kts.)

= 0.083 lb. (16 kts.)

= 0.203 lb. (22 kts.)

$$N_M = \sqrt{\lambda_1} \; N_P \qquad \sqrt{\lambda_1} = \sqrt{120} = 10.95$$

N_M = 444 RPM = 7.4 RPS (10 kts.)

= 532 RPM = 8.86 RPS (12 kts.)

= 711 RPM = 11.85 RPS (14 kts.)

= 830 RPM = 13.84 RPS (18 kts.)

= 954 RPM = 15.9 RPS (20 kts.)

= 1130 RPM = 18.8 RPS (22 kts.)

Tanker (from reference (54))

$LWL = 5.708$ feet

$\Delta = 62.5$ lb.

$S = 6.354$ ft.^{2}

$\frac{1}{2}\rho = 0.970$ slugs/ft.^{3}

$t = 0.14$

$V_{designed} = 1.67$ kts.

V_M(kts.)	V_S(kts.)	$(C_T)_M \times 10^3$
0.992	10.9	5.046
1.450	15.9	5.217
1.688	18.5	5.424
2.009	22.1	6.918

where S = wetted surface

ρ = water density

t = thrust deduction coefficient

C_T = total resistance coefficient

$$C_T = \frac{R_T}{\frac{\rho}{2} S(1.689V)^2} = \frac{T(1 - t)}{\frac{\rho}{2} S(1.689V)^2}$$

or $$T_M = \frac{C_T \frac{\rho}{2} S(1.689V)^2}{1 - t}$$

T_M = 0.101 lb. (10.9 kts.)

= 0.225 lb. (15.9 kts.)

= 0.315 lb. (18.5 kts.)

= 0.57 lb. (22.1 kts.)

The last value of T_M = 0.57 lb. is considered to be above the normal range of power level and will be dropped from consideration.

2. Variations

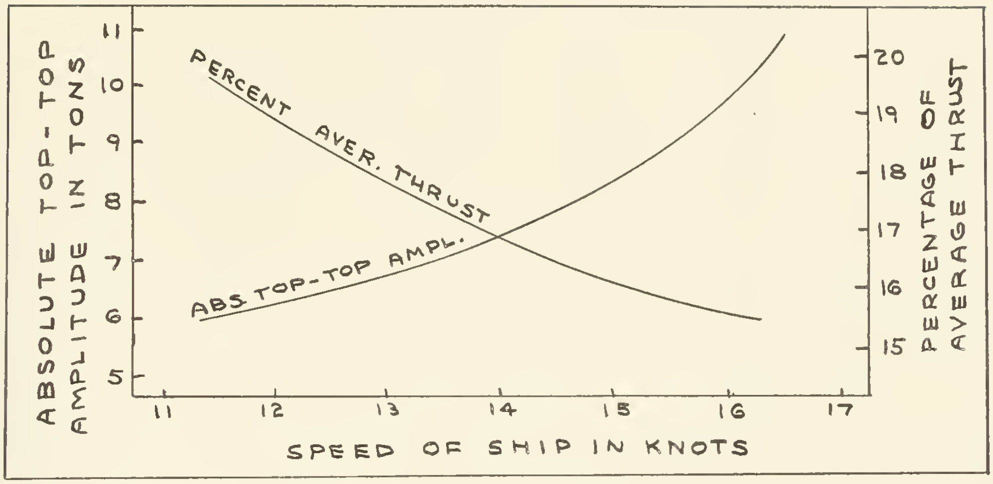
The literature provides results of dynamic measurements of torque and thrust variations. Table IV is intended to be representative of these dynamic measurements and is developed from data presented in reference (55). All the data in Table IV is presented only for the full power condition. Figure 6, taken from reference (56), demonstrates the trend of dynamic measurements as a function of speed. It can be seen that although the amplitudes of force variations expressed as percentages of the mean values decrease with increasing speed, the absolute magnitudes of the amplitudes increase with increasing speed. The values of interest are the smallest absolute variations, thus the values presented in Table IV must be scaled to a lower speed or power level. Assuming that Figure 6 is representative of the relationship between force fluctuations and ship speed,

Table IV

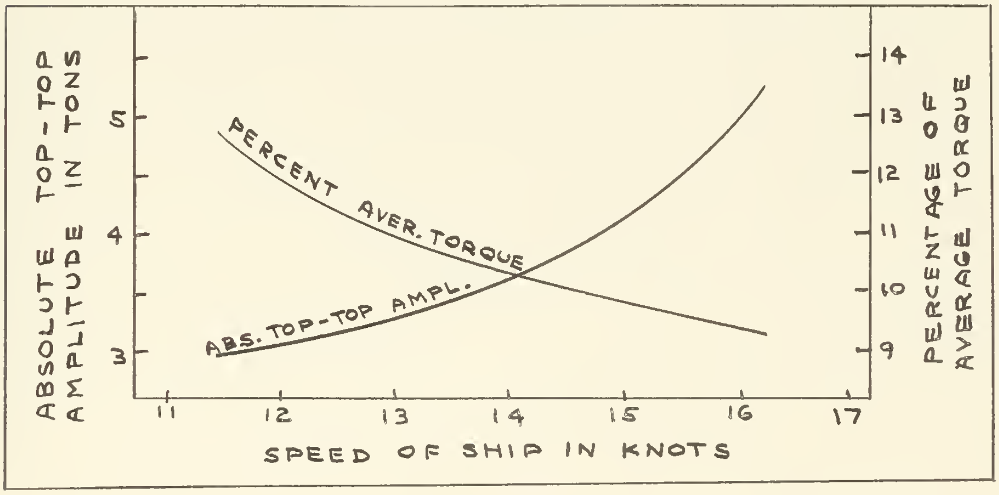
Results of Dynamic Torque and Thrust Measurements

Afterbody	Condition	Power absorption per cent, or speed in kts	No. Prop. Blades	Amplitudes of Harmonic Components as a percentage of Mean Values							
				Torque				Thrust			
				1st	2nd	3rd	4th	1st	2nd	3rd	4th
Mariner	load	100 o/o	4	5.3	1.1	0.6	0.2	9.7	2.7	1.1	0.3
Mariner	load	100 o/o	5	2.0	0.7	0.1	0.2	3.1	2.0	0.2	0.1
Mariner	ballast	100 o/o	4	4.8	1.3	0.5	0.2	8.8	1.9	1.3	1.3
Mariner	ballast	100 o/o	5	3.1	1.2	0.3	0.3	5.2	1.8	0.8	0.0
Conventional	load	100 o/o	4	7.5	2.1	0.7	0.4	13.0	3.8	0.6	0.3
Conventional	load	100 o/o	5	1.1	1.4	0.3	0.2	2.4	2.8	0.5	0.4
Conventional	load	100 o/o	6	4.3	0.8	0.3	0.0	9.0	2.6	1.0	0.1

Values taken from reference (56).



Thrust Fluctuations



Torque Fluctuations

Figure 6. Thrust and Torque Fluctuations of the Propeller as a Function of the Ship Speed

the following procedure will be used to determine amplitude variation at speeds approximately 65 o/o of full speed (assumed to be the lowest speed of interest):

a. Determine the absolute amplitude variation at full speed using the known percentage variation.

b. Find the absolute amplitude variation at reduced speed by multiplying the value found in 1) above by 0.545 ($\frac{6}{11} = \frac{3}{5.5} = 0.545$), the change in torque and thrust variations in going from full speed to a lower speed.

The speed range of interest has been selected to be the same as in Figure 6 , i.e. 65 - 100 o/o full speed. This procedure is admittedly arbitrary and subject to question, but the authors consider it to be sufficient for estimating the smallest force values of interest.

Several conclusions can be drawn from Table IV concerning the harmonic components of the torque and thrust variations:

a. The first harmonic component of 4-bladed and 6-bladed propellers is the most important.

b. The first and second harmonic components of 5-bladed propellers are important.

c. The torque and thrust variations of the 4-bladed propellers are much larger than the variations of 5-bladed propellers.

d. Variations of thrust are larger than variations of torque, percentage-wise.

Representative values of torque and thrust variations, at speeds approximately 65 o/o of full speed, to be used in the Analysis section are:

Torque

4-bladed	5-bladed
0.545 x 4.8 o/o = 2.6 o/o F.S. (first harmonic)	0.545 x 2.0 o/o = 1.1 o/o F.S. (first harmonic)
	0.545 x 0.7 o/o = 0.4 o/o F.S. (second harmonic)

Thrust

4-bladed	5-bladed
0.545 x 8.8 o/o = 4.8 o/o F.S. (first harmonic)	0.545 x 2.4 o/o = 1.3 o/o F.S. (first harmonic)
	0.545 x 1.8 o/o = 1.0 o/o F.S. (second harmonic)

This means that the smallest value of interest of torque is 0.4 o/o of the full scale value, and the smallest value

of interest of thrust is 1.0 o/o of the full scale value over the range 65 to 100 o/o full speed. The second harmonic component of blade rate with a 5-bladed propeller determines the smallest value of interest for both torque and thrust. The range of frequencies of interest is from shaft rate (RPS) to the second harmonic of blade rate for a 5-bladed propeller (2 x 5 x RPS).

Appendix B

Design Procedure for the Stern Tube Bearing

Reference (17) reports the results of a comprehensive program which obtained data not previously reported and which provided new information essential to a more complete analysis and design of journal bearings. Digital computer techniques were used to solve the fundamental equations and the results are presented in charts and tables which permit the determination of friction, film thickness, flow, temperature rise, etc., for bearings having arcs of 60, 120, 180, and 360 degrees for the widely used case of a bearing length to shaft diameter ratio of one. The procedure developed for the design of the stern tube bearing of the small ship model is an application of the information reported in reference (17).

The following nomenclature is used in this design development:

W = load, lb
n = shaft rotational velocity, rps
R = journal (shaft) radius, inches
D = $2R$ = shaft diameter, inches
L = axial length of bearing, inches
P = load per unit projected bearing area = $W/2R$, psi
μ = viscosity in reyns, lb sec/in^2
C = radial clearance, inches
F = friction force on journal, lb.
f = F/W = coefficient of friction
β = angular length of bearing arc, degrees
S = $(R/C)^2\mu n/P$ = bearing characteristic number, or Sommerfeld number
K, K' = constants of proportionality

Chart 3 of reference (17) is reproduced as Figure 7 . The following pertinent calculations are based on this Figure, a fixed, 360 degree bearing configuration, and the fact that model testing will be conducted at a constant shaft rotational velocity.

$$S = (R/C)^2 \mu n/P = (R/C)^2 \mu n 2RL/W = K/W$$

$$\text{Torque loss} = RF = (R/C)f\ (CW)$$

From Figure 7 :

$$(R/C)f = 20S, \quad .4 \leq S$$

$$(R/C)f \cong 20S, \quad .1 \leq S \leq .4$$

Therefore,

$$\text{Torque loss} = 20CWS = K'WS = K'WK/W = K'K$$

Thus, torque loss is constant for a bearing that operates in the range, $S \geq 0.1$. The problem, then, is to design the stern tube bearing to operate in this range of bearing characteristic numbers. The final bearing design will establish values for R, L, and C. The constraints of the design are the range of n for which different model tests will be conducted and the range of W within which shafting system weight and balance must be designed.

Figures 8, 9 and 10 show a plot of the upper limit of bearing load versus shaft rotational velocity for the design

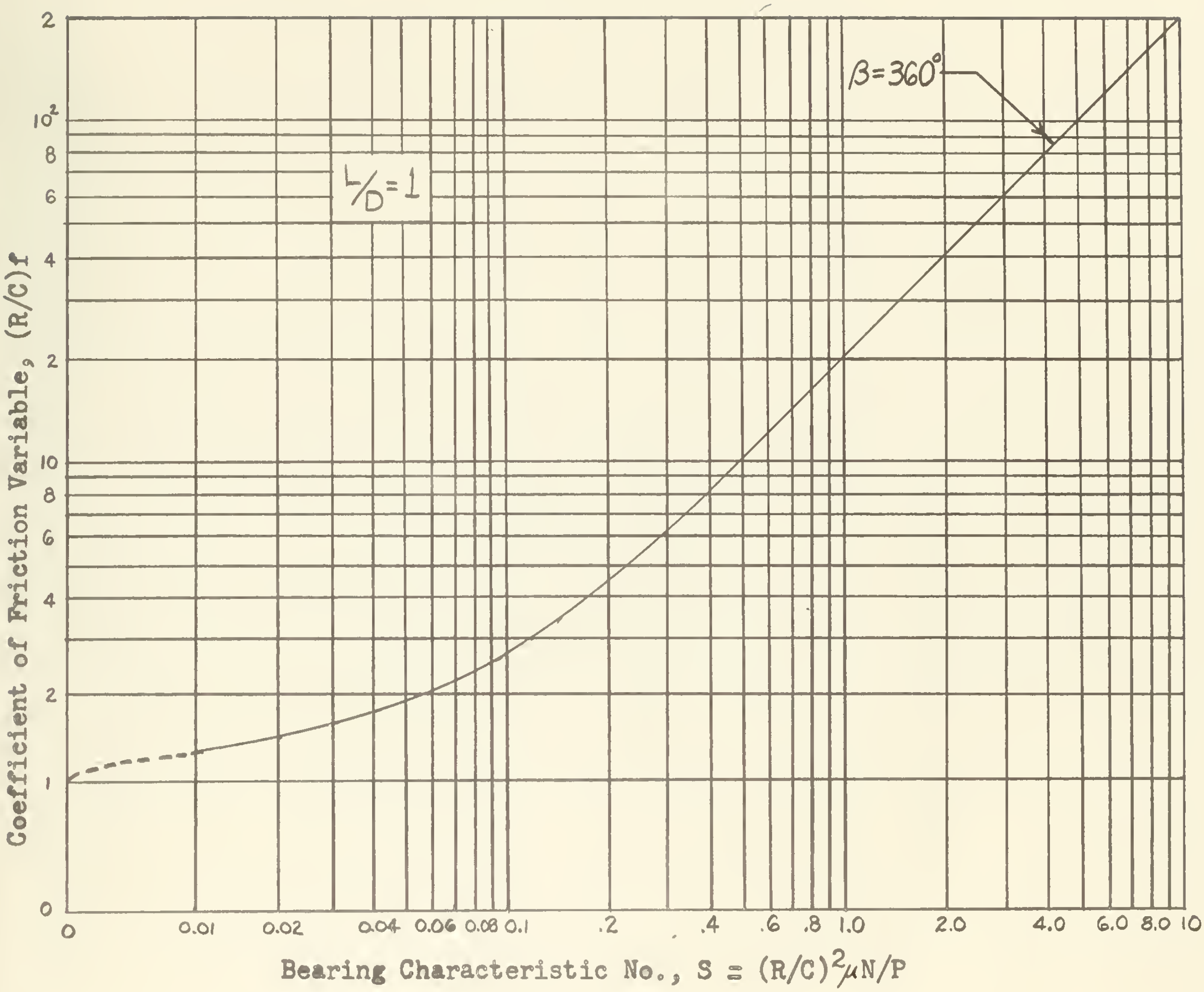


Figure 7. Relationship Between the Coefficient of Friction Variable and Bearing Characteristic Number for Bearing Arc Length of 360°.

parameters, shaft radius and radial clearance. The following equations were used to prepare these figures:

$$S = (R/C)^2 \mu n/P = (R/C)^2 \mu n 2RL/W = 4\mu nR^4/C^2W$$

Solving for W,

$$W = 4\mu nR^4/C^2S$$

Substituting $S = 0.1$,

$$W_u = \text{upper limit bearing load} = 40\mu nR^4/C^2$$

$$\mu = 2 \times 10^{-5} \text{lb. sec/ft}^2 = 1.39 \times 10^{-7} \text{ lb. sec/in}^2$$

for fresh water at 70 degrees F.

The range of interest for n was previously determined:

$$13 \leq n \leq 20 \text{ rps.}$$

Model size limits the range of interest for R:

$$0.125 \leq R \leq 0.1875 \text{ inches}$$

Common practice places the range of interest for C:

$$0.00025 \leq C \leq 0.001.$$

Shaft radius and radial clearance should be chosen such that the allowed range of bearing load, W, is as large as possible. Thus, any bearing load variations which cannot be eliminated are still within the range for a constant torque loss. The choice, however, is not so simple as it seems. Although the range of allowed bearing load increases in

proportion to R^4 while shaft weight only increases in proportion to R^2, the mass of the shaft system must be compatible with the overall mass allowed by the frequency analysis. The frequency requirements are discussed in the Measurement System portion of the Analysis Section.

It is the opinion of the authors that Teflon would be a good bearing material for the stern tube. The use of Teflon, however, and of plastics in general, for bearing material should be investigated further prior to application to the small ship model. Reference (57) should be consulted for this investigation.

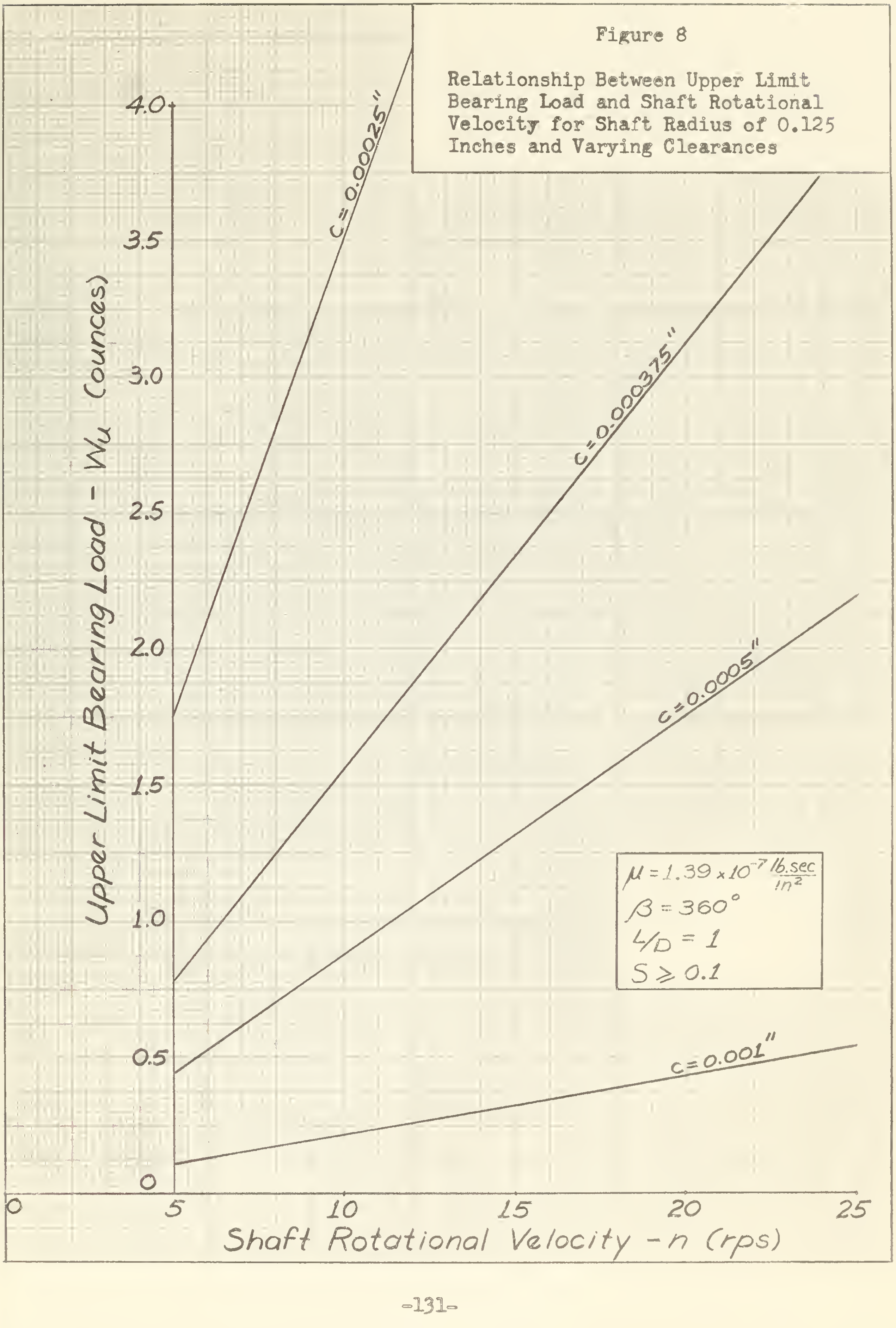


Figure 8

Relationship Between Upper Limit Bearing Load and Shaft Rotational Velocity for Shaft Radius of 0.125 Inches and Varying Clearances

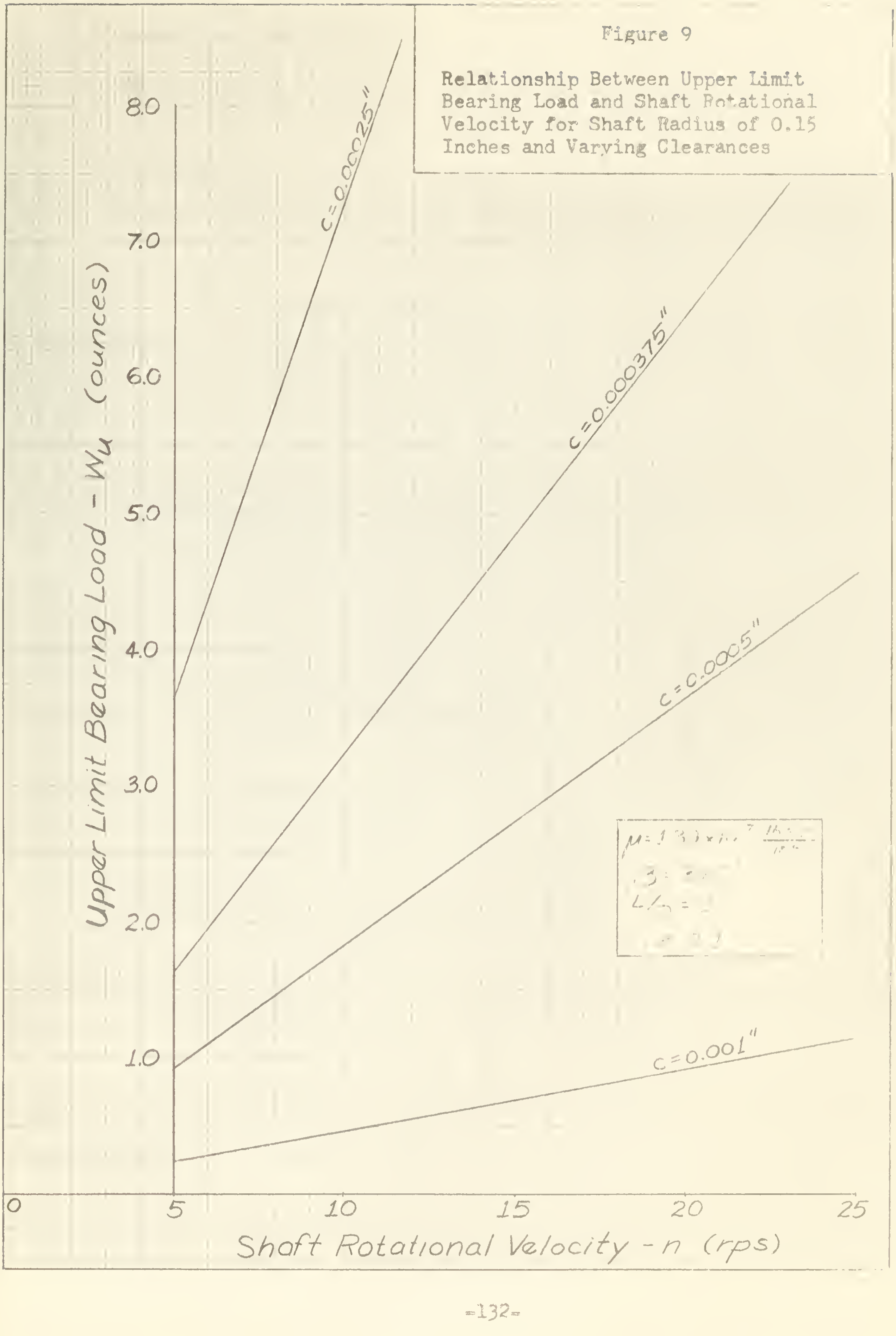


Figure 9

Relationship Between Upper Limit Bearing Load and Shaft Rotational Velocity for Shaft Radius of 0.15 Inches and Varying Clearances

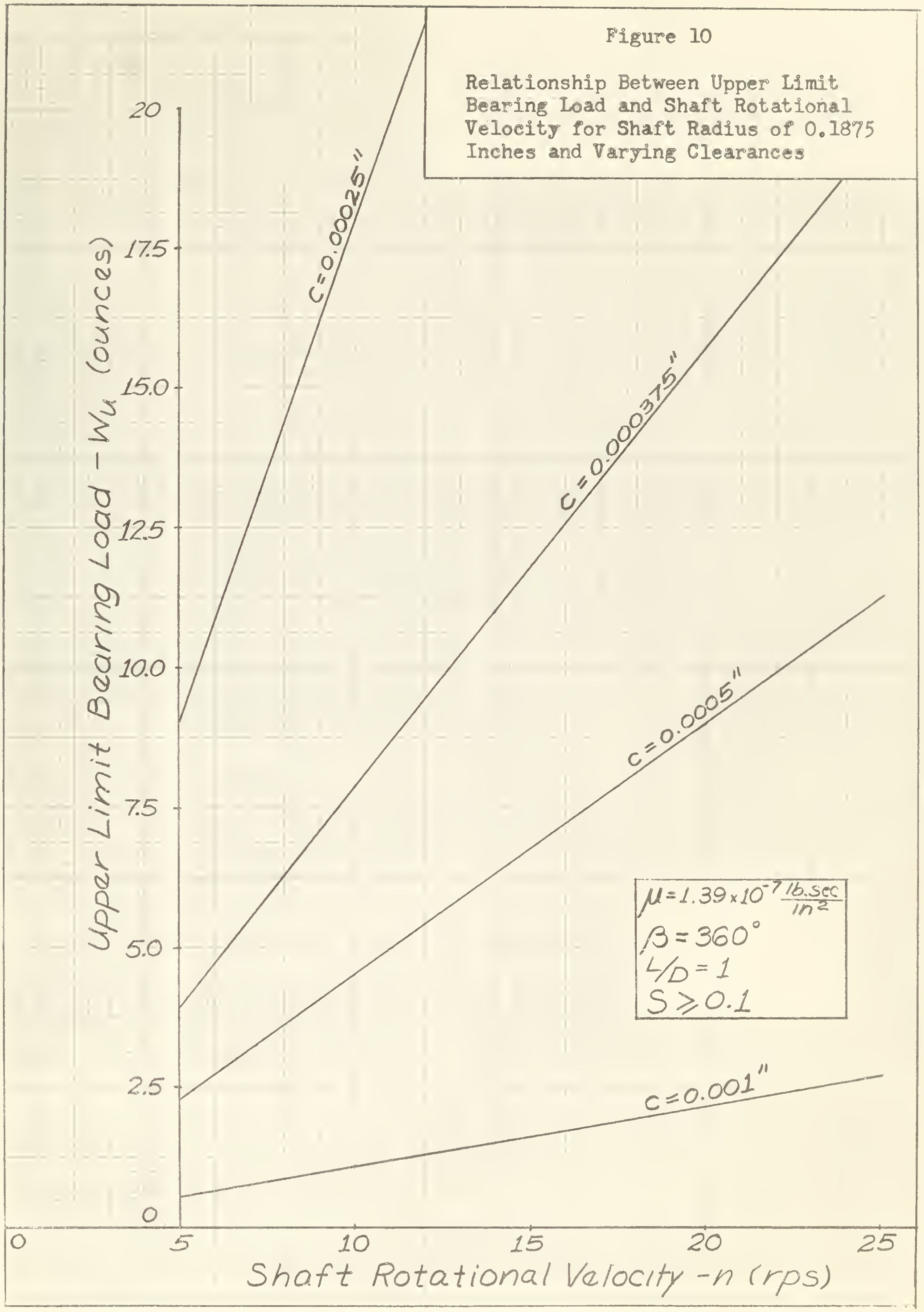


Figure 10

Relationship Between Upper Limit Bearing Load and Shaft Rotational Velocity for Shaft Radius of 0.1875 Inches and Varying Clearances

Appendix C

Procedures for the Determination of Measured Strain and Other Measurement System Parameters

This Appendix develops several parameters for use in the dynamic analysis required for the actuating device. It also presents suggested procedures for the determination of measured strain resulting from a particular measurement system configuration. Since the analytical models developed in this Appendix are those used for the displacement-sensitive actuating devices, much of this discussion applies to both strain and displacement types.

1. Determination of model propeller characteristics

The model propeller weight, $(W_p)_M$, can be approximated by an equation presented in reference (58):

$$W_p = K D_p^3 (MWR)(BTF)$$

where

$k = 0.20$ for 3 bladded propellers (bronze)

$= 0.26$ for 4 bladed propellers (bronze)

D_p = propeller diameter

MWR = mean width ratio

BTF = blade thickness fraction

By appropriate scaling from a Mariner parent with $L = 520$ feet, $D_p = 22$ feet, $L_M = 5$ feet:

$$(D_p)_M = \frac{5 \times 22}{520} = 0.2115 \text{ feet} = 2.54 \text{ inches}$$

Assuming values of MWR = 0.4 and BTF = 0.05,

$$(W_p)_M = (0.26)(2.54)^3(0.4)(0.05)$$

$$= 1.05 \text{ ounce (Schoenherr propeller)(bronze)}$$

This is only an approximation and could be reduced by selection of a lighter material such as aluminum.

$$(W_p)_{\text{Aluminum}} = 0.31\ (W_p)_{\text{Bronze}}$$

The model propeller moment of inertia, $(I_p)_M$, can be obtained from the expression:

$$I_p = mk^2$$

where m = propeller mass

k = radius of gyration

$= (0.2 - 0.22)D_p$

Assuming $(W_p)_M = 1.0$ oz. $= 0.0625$ lb.,

$$(I_p)_M = \frac{(0.0625)(0.22 \times 2.54)^2}{386}$$

$$= 4.91 \times 10^{-5} \text{lb.in.sec.}^2 \text{(bronze Schoenherr propeller)}$$

This inertia is then increased by 25 o/o to account for entrained water.

$$(I_p)_M = 613 \times 10^{-7} \text{lb.in.sec.}^2 \text{ (bronze Schoenherr propeller)}$$
$$= 287 \times 10^{-7} \text{lb.in.sec.}^2 \text{ (aluminum Schoenherr propeller)}$$

2. Suggested procedure for the determination of measured strain

   a. For thrust measurement

Having selected a desired thrust actuating device natural frequency from dynamic considerations, a value for thrust system mass is assumed. This mass will include the mass of the propeller and hub, entrained water, some shafting, and possibly some instrumentation (depending upon the relation of the thrust sensing and torque sensing systems on the propeller shafting). Equation (1) is solved for the allowable actuating device stiffness, k_1.

$$\omega_m^2 = \frac{k_1}{m} \qquad (1)$$

Assuming that the actuating device configuration is composed of four mutually perpendicular cantilever beams, a simple cantilever analysis is made using the deflection equation:

$$x = \frac{PL^3}{3EI}$$

where x is the deflection of the end of the beam, P is the applied load ($\frac{T}{4}$ for this case), E is Young's modulus, and I is the moment of inertia of the beam cross-section about its mid-depth.

Substitution of $P = \frac{T}{4}$ and rearranging gives:

$$1/k_1 = \frac{L^3}{12EI}$$

The measurable strain value is obtained from the relation:

$$\epsilon = \frac{TLc}{4EI}$$

where c is the distance from the beam neutral axis to the outermost fiber, or the beam half depth.

The values of L, E, I, and c are then selected in such a way as to maximize the value of strain. There are four major constraints placed upon the configuration: the stiffness of the shafting must be much greater than the stiffness of the actuating device so that most of the deflection of the system occurs in the actuating device; the natural frequency of the cantilevers must be at least ten times the highest frequency of interest; the cantilevers must have sufficient space to allow for the proper mounting of strain gages; and the maximum cantilever length is limited to 0.5 to 1.0 inch by the cleared space available within the model hull around the propeller shaft.

The mass resulting from this procedure is then compared to the mass assumed for the solution of equation (1). This procedure is repeated until the assumed value and calculated value of mass, m, are in agreement. The value of strain obtained from this procedure is an idealized value, and usually will not be the value of strain detected by the strain gage. This must be recognized in the design stage of the dynamometer development.

A sample calculation is made using the above procedure to indicate a typical value of strain resulting from a cantilever thrust actuating device configuration:

Assuming: $\omega_m = 2000$ cps

$m = 2.5/386$ oz. (mass)

$L = 1$ in.

$E = 10 \times 10^6$ psi (aluminum)

$T_1 = 0.32$ lb. (highest value of interest)

$T_2 = 0.0032$ lb. (lowest value of interest)

$$k_1 = m\,\omega_m^2 = \frac{2.5(2\pi\ 2000)^2}{(16)(386)}$$

$$= 63{,}900 \text{ lb./in.}$$

$$I = \frac{k_1 L^3}{12E} = \frac{(63{,}900)(1)}{(12)(10 \times 10^6)}$$

$$= 0.000532 \text{ in.}^4$$

$$I = \frac{1}{12}\, b\, h^3 \text{(for rectangular beam)}$$

where b = cantilever breadth

h = cantilever depth

Assume minimum b = 0.25 in. (for mounting strain gage)

$$0.000532 = \frac{1}{12}(0.25)h^3$$

$$h^3 = \frac{12(0.000532)}{0.25}$$

$$= 0.0255 \text{ in.}$$

$$h = 0.294 \text{ in.}$$

$$c = \frac{h}{2}$$

$$= 0.147 \text{ in.}$$

$$\epsilon_1 = \frac{TLc}{4EI}$$

$$= \frac{(0.32)(1)(0.147)}{(4)(10 \times 10^6)(0.000532)}$$

$$= 2.21 \text{ microin./in. (microstrain)}$$

$$\epsilon_2 = 0.0221 \text{ microin./in. (microstrain)}$$

This configuration would then be refined and optimized as shown, but these values of strain are somewhat representative of the magnitudes produced.

The use of a thin-walled cylindrical actuating device for the production of axial strain is not developed. In general, higher strains can be produced by a beam arrangement similar to the type analyzed.

b. For torque measurement

The procedure for determining the values of strain resulting from a particular configuration to measure torque is similar to that for thrust. For the case of torque, however, the selection of an appropriate dynamic analytical model is more complex. For the preliminary analysis, a two mass system connected by a spring element may be used. This spring, or actuating device, actually occupies a fractional portion of the length of shafting connecting to the two masses in the analytical model. This point will be clarified as the analysis progresses.

The desired actuating device natural frequency has been determined as in the thrust measurement procedure. The moments of inertia for the two end masses of the analytical model are then assumed. One mass, I_1, consists of the propeller and hub, entrained water, some shafting, and possibly the mass of the thrust measuring actuating device and transducer (dependent upon the fore and aft arrangement of the torque and thrust systems). The other mass consists of some shafting, possibly the torque transducer and associated equipment, and a flywheel. The flywheel is necessary to

damp out any drive motor noise or other vibrations discussed in the Input section of this thesis. Equation (2) is then solved for the allowable actuating device stiffness, k_2.

$$\omega_m^2 = \frac{k_2(I_1 + I_2)}{I_1 I_2} \qquad (2)$$

This value of stiffness is then introduced into the following equation which relates the various actuating device parameters:

$$k_1 = \frac{GI_p}{L} = \frac{E I_p}{2(1 + \mu)L}$$

where G is the actuating device modulus of rigidity, I_p is the device polar moment of inertia, L is the device length and μ is the Poisson's ratio for the device material.

Strain, γ, is determined by the expression:

$$\gamma = \frac{QR}{I_p G}$$

where Q is the torque value of interest and R is the radius of the actuating device to the outermost fibers.

The obvious way to produce the maximum strain is to select a material with the smallest modulus of rigidity

and a configuration with the largest radius and length, but the smallest polar moment of inertia. The following constraints, however, limit the extent to which this may be done: the actuating device will experience creep as did the Saran device of reference (8); the increase of R and a decrease of I_p, acting contrary to each other, requires the use of a thin-walled section which may fail under applied loads; the maximum radius is limited to the same 0.5 to 1.0 inch clearance space that limited the thrust device; the increase of length increases the possibility of shafting whirling or other radial deflection; and the portion of the shaft not included in the actuating device must be much stiffer than the actuating device so that the actuating device deflects to its maximum extent under the applied loads.

Consideration should be given to the use of a cantilever-type device similar to that discussed for the thrust case, but with the flexural deflection taking place in a torsional direction. The equations for the analysis of this type device are presented in reference (12).

Appendix D

Principles of Transducer Operation

The four main groups of mechano-electrical transducer elements are: variable parameter analog, self-generating analog, frequency or pulse generating, and digital. The principles of operation of the types of elements within each group are briefly discussed in this Appendix. The descriptions are basically those of the transducer references (13, 22, 27, 38, 40, 59). These references include detailed descriptions and performance characteristics and should be consulted where additional information is desired. Specific references are given where the treatment of the element is of particular note.

1. Variable parameter analog

The variable parameter analog transducers produce an output which is a proportional fraction of an original resistance, capacitance, or inductance. The changes in resistance, inductance, or capacitance caused by the motion of the actuating device are translated into corresponding voltage or current effects.

a. Variable resistance transducers

The resistance of a potentiometer is varied by sliding an arm across a number of wire turns. Potentiometers are

not considered applicable for measurements in the small ship model because resolution is limited by the number of wire turns and their uniformity from turn to turn. Their response to higher frequency measurements is also restricted due to the inertia of the arm and the tendency of the sliding arm to bounce and break contact.

Strain gage transducers are of two types: bonded and unbonded. The bonded type incorporates single-bonded wire gages, metal foil gages, or semiconductor gages. The wire gage operates on the principle that its length and diameter are altered, when it is stretched elastically, resulting in an overall change of resistance. The metal foil gage utilizes a strain sensitive grid. Its significant advantages include improved hysteresis, fatigue life, and lateral strain sensitivity, in addition to improved transmission of strain from the test surface to the gage. Semiconductor strain gages use piezoresistive solid-state materials as strain elements to provide sensitivity almost two orders of magnitude greater than wire or foil gages. Temperature sensitivity is their major disadvantage.

Bonded strain gage transducers are rugged, relatively simple and have excellent linearity and hysteresis characteristics.

Sufficient actuating energy must be available for good results because of low sensitivity in the wire and foil gages and limited lower range performance. Use of the semiconductor gages is more promising, however, because of the increased sensitivity.

The unbonded type of strain-gage transducer consists of a stationary frame which supports a movable armature. The strain sensitive wire is strung under initial tension as a filament between pins located on the frame and armature. As the armature is displaced by the external force, the strain increases in one pair of filaments and decreases in the other pair. Unbonded strain-gage transducers are used most advantageously in the measurement of low magnitude forces.

Strain-gage transducers generally possess excellent frequency response in both the high and low ranges. They can be made quite small, less than one cubic inch, and can be sealed for use underwater.

b. Variable capacitance transducers

Capacitance is a function of effective area of the conductors, separation between them, and the dielectric strength of the material in the separation. Variation

of any one of these three parameters causes a change in capacitance. The change in capacitance with respect to plate separation is greater than changes with respect to area or dielectric strength. The advantages of capacitance transducers are small size, excellent high-frequency response, high temperature resistance, good linearity, good resolution, high sensitivity to small displacements, and ability to measure both static and dynamic quantities. Major shortcomings are sensitivity to temperature, high impedance output and the comparative complexity of associated electronic equipment. Capacitance transducers must be reactively as well as resistively balanced.

c. Variable inductance transducers

Variable inductance transducers use the magnetic properties of core materials and air gaps to alter the self-inductance of fixed coils. When the reluctance of a magnetic circuit is altered, both the inductance and the inductive reactance of coils in the circuit will be similarly changed. This principle is used when the air gap in the magnetic core or the permeability of the core material itself are varied in the transducer. Transducers

that use the air gap change are known as variable reluctance types while those using a variable core permeability to change reluctance are known as variable permeance types.

A descriptive ship model application of the variable reluctance principle is given in reference (6). The transducers described use a differential arrangement whereby one gap is increased as the other is decreased. Such a differential arrangement has greater sensitivity than a single element. An application of the principle to the small ship model is given in reference (12).

The variable inductance transducing element is used most widely in displacement-measuring transducers with relatively low natural frequencies, i.e., transducers with usable frequencies below approximately 100 cps.

d. Variable transformer transducers

In the variable transformer the mutual inductance between two coils and a magnetic field is varied, either by a change in the position of the field itself or by a change in the relative position of the coils. Output voltage is induced in a secondary winding and varied by the position of the primary. The change in position may be longitudinal or rotational. The most common examples are the differential transformer and the Selsyn systems.

The typical differential transformer consists of a hollow concentric nonmagnetic form, on which are mounted three windings -- one primary and two secondary. The position of a core placed within the coil form determines the relative mutual coupling, and thus voltage developed, between the primary winding and each of the secondary or output windings. An excellent description of linear differential transformers is given in reference (35).

Selsyn systems are motorlike devices used to form rotary position sensing and indicating systems. Many specialized, small angle, high precision transducers use one or more of these principles. These transducers are known variously as Magnesyns, Inductosyns, Synchrotels, and Microsyns and are described in detail in reference (44).

2. Self-generating analog

Self-generating analog transducers require no outside source of power and often generate sufficient voltage or current to require only simple measuring circuitry. These transducers may be divided into the following types: piezoelectric, photoelectric, electrokinetic, and electron-tube.

a. Piezoelectric transducers

Piezoelectric transducers use the principle that a force or stress, applied along specific planes of certain crystalline dielectrics, produces a relative displacement of charges within the lattice and thus generates a voltage across the crystal. A synthetic ceramic, barium titanate, is free from limitations imposed by crystal structure; thus, by molding it into different shapes and sizes, an electrical axis can be "built in". The piezoelectric transducer is simple and rugged, suitable for many applications where sufficient mechanical force is available. Its greatest disadvantage is its lack of response to steady-state displacements or forces.

b. Photoelectric transducers

Photoelectric transducers depend upon changes in light energy for their operation and seldom load or interfere with the phenomenon being measured. Sensitivity and speed of response of these transducing elements are high. When coupled with appropriate optical systems, light sources, and amplifiers, they may be used for a wide variety of physical measurements. They are classified as photoemissive, photovoltaic, and photoresistive.

The photoemissive sensor, or phototube, derives its electron flow from a cold cathode which emits electrons when light falls upon its surface. An external d-c power supply is required and a high resistance is placed in the cathode circuit so that the current changes may be sensed as a voltage drop across it for amplification.

Photovoltaic cells require no external source of power; incident light striking the interface of two metal surfaces causes direct generation of voltage. The voltage generated, however, is in the order of low millivolts, the frequency response is relatively poor, and the output is not a linear function of light intensity.

The photoconductive cell and the photodiode are much smaller, more sensitive, and more rugged than any of the other photoelectric transducers. The photoconductive cell varies its electrical conductivity in accordance with the light intensity it receives. The photodiode and the phototransistor are semiconductor devices operating on the principle that light gives enough energy to the valence-bond electrons to raise them to the conduction band. The phototransistor is quite small, some being no larger than

a matchhead; they are adapted for applications where a large number of individual, light-sensitive elements are required in a small area.

c. Electrokinetic transducers

The electrokinetic principle utilizes the electrical signal generated by a minute flow of polar fluid passing through a microporous ceramic disk. The potential difference produced between the opposite faces of this disk is proportional to the applied pressure. Transducers using this principle are inherently dynamic instruments designed to measure periodic phenomenon only. This type of transducer is valuable in the measurement of high intensity sound and noise. The disadvantages of electrokinetic transducers are their inability to measure static conditions, poor zero-restoring properties, and diaphragm distortion by electrical bias.

d. Electron-tube transducers

Electron-tube transducers employ an electron tube, usually subminiature, with an element or elements free to move. The output of the tube varies in proportion to the displacement of element position. Force may be measured

using suitable linking mechanisms to change the original quantity into a displacement.

High sensitivities can be obtained with a triode system such as is used in RCA's tube type 5734. Used with a load resistor of about 50,000 ohms and a plate supply of 20 volts, the displacement-voltage sensitivity of the system is in the region of 5 millivolts per microinch of displacement.

An ionization transducer uses a glass tube filled with gas under pressure and containing two electrodes. When this tube is brought into an electric high-frequency field between movable plates of a capacitor, a voltage proportional to the displacement is induced. Sensitivity can reach values of 50 to 100 millivolts per microinch of displacement.

The prime advantage of these transducers lies in the high frequency response of the large output signals that are produced with very small mechanical displacement. They are temperature sensitive, however, and must be used under controlled conditions.

3. Frequency or pulse generating

Frequency and pulse-generating transducers measure physical variables in terms of a pulse repetition rate or frequency signal rather than in the form of continuously varying currents or voltages. Because frequency is an analog quantity and depending on how their output information is handled, these transducers may be considered as either analog or digital devices. Included in this group are frequency modulating, frequency generating, and pulse counter transducers.

a. Frequency modulated transducers

The two principal forms of FM transducers are the vibrating wire and the variable reluctance gage and oscillator.

The vibrating wire transducer contains a stretched wire main element which is placed in a magnetic field and maintained in vibration at its natural frequency. Upon application of a force to the actuating device, the tension in the wire changes. In this manner the mechanical displacement is converted into a change in frequency which is inversely proportional to a positive displacement.

The reluctance gage and oscillator includes a tiny oscillator circuit as an integral part of the transducer

housing. A change in reluctance, thus inductance, of the unit, produces a proportional change in the output frequency of the oscillator. The capacitance in the oscillator circuit could also be changed by displacement, but this type is not so common.

b. Frequency-generating transducers

Frequency-generating transducers produce a simple series of voltage or current pulses, or cycles in proportion to the change in the physical parameter being measured. In particular, pulse-generating pickups operating with the principles of variable reluctance or the phototube and light source combination are most reliable.

The electromagnetic reluctance pulse pickup is a small permanent magnet on which is wound a coil. The field of the magnet is varied momentarily by the motion of an external magnetic body passing near it. A voltage pulse is generated at the coil because of the change in flux surrounding the coil. This principle is often used to measure shaft speed.

The phototube may be used to detect rotation and other motions either by the interruption of the light beam or by the detection of translucent and opaque portions of moving

components. Methods used include sensing the reflection of a white spot on a rotating shaft and sensing light beam interruptions caused by a perforated wheel. The series of discrete pulses produced are counted over a precise time interval.

4. Digital encoders

Unlike analog transducers which are characterized by proportional measurement, or pulse rate transducers, characterized by proportional counting, digital encoders and encoder transducers produce a distinct coded output signal. As the physical variable changes, the incremental state of the output coded signal changes in such a manner as to represent the new value in the form of a nonambiguous coded value. Time sampling is not involved and the transducer output may be scanned at any desired rate to readout its new value. Digital encoders produce a true digital output suitable for direct entry into digital computers or data handling systems without further conversion.

The shaft position encoder is the most efficient of the digital encoders. It consists of a shaft attached to a disk mask or a drum with a digital coded scale. The scale may be

formed with either a combination of conducting and nonconducting areas of a combination of translucent and opaque surfaces. There exists a definite coded form for each discrete position of rotation of the disk. Resolutions are from 100 points, for a two inch diameter disk, to 50,000, for ten inch disk, unique positions per 360° turn. They are readout by either a series of brushes or a photocell arrangement.

Rectilinear encoders are linear scale transducers which have the same basic advantages as the shaft position encoders described above. These devices utilize the same principles as the rotary type except that the scale is made over a linear range and motion is a rectilinear displacement which produces the coded output presentation.

Appendix E

Description of Signal Conditioners

The types of signal conditioners described in this Appendix are classed as follows: input modification and instrumentation amplification. The descriptions are basic and no attempt is made to describe all types of signal conditioners. These descriptions are taken from the transducer references (13,22,27,3[illegible],40,59) which should be consulted where information in addition to that given here and in the Analysis is desired

1. Input modification

Input modification may accomplish any of the following: conversion of the output into a voltage, current, or digital code; straightforward amplification of the transducer output; filtering out of unwanted frequencies from both transducers and associated circuitry; and impedance matching or signal attenuation. The most common circuit forms used are: simple current-sensitive circuits, ballast circuits, balanced bridge circuits, resonant circuits, signal preamplifiers, and attenuators and filters.

a. Simple current-sensitive circuit

A simple current-sensitive circuit employs any one of the various forms of variable resistance elements. The transducer is placed in series with a voltage source and a current indicator or recorder which senses output current. The current variations are caused by the variable resistance.

b. Ballast circuit

The ballast circuit is a variation of the current-sensitive circuit. Instead of a current-sensitive indicator or recorder through which the total current flows, a voltage-sensitive device (some form of voltmeter), is placed across the transducer. Now the voltage variations caused by the variable resistance are sensed.

c. Balanced bridge inputs

The most common single transducer circuit configuration is the Wheatstone resistance bridge and its a-c counterparts for inductance and capacitance. Strain gages and transducers incorporating them are almost always connected in this configuration. Similarly, many other types of sensors are used as one or more active arms of the four-arm bridge circuit.

With bridges employing either capacitive or inductive arms, only a-c excitation is used. Voltage regulation is critical because of the effect on output due to a change in excitation voltage.

d. Resonant Circuits

Capacitance-inductance combinations present varying impedance, depending on their relative values and the frequency of the applied voltage. When connected in parallel, the inductance offers small opposition to current flow at low frequencies, while the capacitive reactance is low at high frequencies. At some intermediate frequency, the opposition to current flow, or impedance, of the combination is a maximum. The frequency corresponding to this maximum effect is known as the resonant frequency. If a capacitive transducer is used, it could be employed in combination with an inductive element to form a resonant combination. Variation in capacitance caused by variation in the input signal would then alter the resonant frequency, which could then be used as a measure of the input.

e. Signal preamplifiers

Preamplifiers and attenuators are used for signal level changes, equalization, or impedance matching. Transducers with high impedance outputs require special treatment in order to reduce the errors caused by poor impedance matching. Electrometers, input transformers, cathode followers, transistor emitter followers, and other circuits are used as isolation or impedance matching devices. Preamplifiers and filters may be used to extend or reduce the frequency range of a transducer or remove unwanted frequencies from the transducer signal.

f. Attenuators and filters

To normalize or scale down the amplitudes of signals, or to eliminate unwanted frequency components, attenuators, and filters respectively are used. Attenuators used in instrumentation take many forms. For a-c circuits above 5 kc, compensated attenuation circuits are required to maintain constant phase shift and provide high frequency equalization. Filters are used in data instrumentation to suppress unwanted frequency components, select bands of frequencies for separate recording or indication, and remove the harmonic content of inductance bridge output

and carrier frequency components in carrier or chopper demodulation. They are used to remove hum, ripple, and noise in power supplies used for bridge excitation and to power electronic measuring equipment. The simplest filter circuits consist of series or shunt resistances and capacitances.

2. Instrumentation amplification

The following is primarily a discussion of applications of instrumentation amplification rather than a description of the types. The following categories are discussed: voltage and power amplifiers, a-c and d-c amplifiers, chopper amplifiers, and carrier amplifiers.

a. Voltage and power amplifiers

Amplifiers used in data instrumentation are of both the voltage and power types. A voltage amplifier may be employed when in this case, an indicator such as a cathode-ray oscilloscope or a vacuum-tube voltmeter might be used at the readout device. If a recorder or some form of controller must be driven, then a power amplifier would be necessary to boost the energy available to drive the device.

b. a-c and d-c amplifiers

Amplifiers may also be classified according to the character of input signal which they will accept. Reliable amplifiers that will amplify d-c inputs are difficult to design. A constant d-c amplification or gain is difficult to maintain. As a result, a-c amplifiers are used wherever possible. The a-c amplifier will accept only varying inputs, whereas the d-c amplifier will amplify constant as well as varying inputs. A d-c voltage amplifier therefore will amplify such things as the voltage from a battery as well as a varying voltage, while the a-c amplifier will ignore a d-c voltage and amplify only any varying components.

c. Chopper amplifiers

A simple a-c amplifier may be used to amplify a d-c input through use of an additional circuit component known as a chopper. The chopper is an electrically driven switch often driven at either 60 or 100 cps, although a speed of 400 cps is sometimes used. Application of an alternating voltage to the driver coil causes a reed to vibrate between

a pair of contacts. When a d-c signal input is connected, the amplifier receives a chopped or square-wave voltage which, being an a-c amplifier, it amplifies without trouble.

d. Carrier amplifiers

When the input signal information is carried on an a-c frequency, the input signal is said to modulate the carrier frequency. The output from a differential transformer is an example of a modulated signal. The transformer is energized by an a-c exciting frequency. Core position, either static or varying, determines the amplitude of the output.

Certain special-purpose amplifiers incorporating the carrier source as a part of the amplifier are referred to as carrier amplifiers. Simple carrier systems provide an a-c output whose amplitude is blind to the sign of the input signal. The phase relation between the power souce and the output does depend on the sign of the input and may be used to determine it. Phase-sensitive arrangements employed to accomplish this are referred to as phase-discriminator or mixer-demodulator circuitry.

Appendix F

Methods for removing signals from a rotating shaft

Signals may be removed from a rotating shaft in at least four ways: by direct connection, by telemetering, by use of sliprings, and by inductive, or magnetic, coupling.

1. Direct connection

When a shaft rotates slowly enough and the data sampling period is not too long, direct connections may be made between the transducer and the remainder of the measurement system. Sufficient lead is provided and the cable is permitted to wrap itself on or off the shaft. If the shaft cannot be stopped quickly enough at the end of a test run, a fast or automatic disconnecting arrangement could be provided. Shielded cable should be used to minimize reactive effects resulting from the coil of cable on the shaft. Although this technique is somewhat limited, it should not be overlooked as a possible means of removing information.

One possible configuration that the authors suggest be investigated is with the wire wrapped on a small diameter extension of the shaft near the motor. Another larger drum, or take-up reel, could be located on the towing tank carriage. This take-up reel may also carry an amplifier which rotates with the drum. This amplified signal could then be transmitted through sliprings from the take-up reel to the remainder of the signal conditioners. The weight of the cable on the shaft does not add to the mass of the actuating device, but simply adds to the desired flywheel effect at the motor end of the shaft.

2. Telemetering

Telemetering is the process of actually transmitting the information through the use of a radio-frequency transmitter mounted on the shaft, and picking up the signal by a receiver placed nearby. This method has been used successfully on large rotating shafts (10), and equipment has been perfected for more general and small shaft use by at least one manufacturer (11). The cost of such a system often makes the arrangement impractical for general use.

3. Sliprings

The most common method for removal of information from rotating shafts is through sliprings. In the small ship model application, the signal voltages may be in the order of millivolts, or fractions of millivolts. Resistance variations within the slipring-brush combination can be as great as or greater than the variation of the resistance in a strain gage, and extraneous voltages can be generated within the sliding contact which will swamp the signal, or perhaps give the same appearance as the signal itself.

References (12) and (13) present the most complete and comprehensive discussions of slipring instrumentation problems, applications, and commercial information that the authors were able to locate. No attempt will be made in this thesis to cover this same material; however, there are certain facts which merit discussion.

Brush noise is classified by Motsinger (12) as follows:

a. Brush bounce

Brush bounce is a circuit interruption that occurs when adverse acceleration forces cause the brush and ring

to part company periodically during the rotation cycle. This is the most serious noise component because the disturbance can be in the order of volts.

b. Resistance modulation

Resistance modulation describes the inconsistent electrical resistance at the brush-slipring interface. The steady-state portion of the resistance (running about 2 ohms) is of little consequence to the system, but a variation of one-tenth of an ohm is equivalent to 500 microstrain. For the small ship model application it is desired to distinguish strain amplitudes ranging from about 2.2 microstrain (highest value of interest) to about 0.022 microstrain (lowest value of interest).

c. Thermal EMF's

Thermal electromotive forces are created by the heat of friction and the dissimilar materials in contact. A virtual network of thermocouples are present, generating random signals in the micro-volt region. Regardless of workmanship and choice of circuitry the minimum noise level will be determined by these thermal emf's.

To better acquaint the authors with current state-of-the-art type sliprings, several visits to the M.I.T. Instrumentation Laboratory were made. It was determined that the MOD J 16 Pendulous Integrating Gyro Accelerometer under development utilizes a low noise slipring with a maximum acceptance noise level limit of 10 microvolts per milliampere, peak-to-peak. This slipring is designed for use at slow revolutions only, however in the order of 3-4 RPM. The size, configuration, and noise specifications for this slipring may make it acceptable for use in the small ship model; however, more information is needed on its wear properties and noise levels at higher rotational speeds.

The noise specifications given in reference (4) appear to be at the state of the art in precision sliprings. This is an advertisement for a slipring with a noise level claimed to be only 3 microvolts per milliampere, peak-to-peak (3mΩ) with rotational speeds up to 3000 RPM and with a life exceeding 1000 hours. The cost of either of these sliprings (not known to the authors) may prove to be prohibitive for use in the M.I.T. facilities.

The use of mercury sliprings is attractive because they eliminate brush bounce. Noise levels below 20 microvolts are possible (62). They were used in an unsuccessful attempt to construct a sensitive dynamometer by Chapman (11). This, however, does not eliminate their potential.

The experience of the previous theses groups (8,9,10,11,12) should prove useful in any possible future attempts to incorporate sliprings in the design of a small ship dynamometer.

4. Inductive coupling methods

The main distinction between telemetry and inductive, or magnetic, coupling as alternatives to sliprings is that in the former case the oscillator rotates, while in the latter case the carrier excitation source is stationary. Several interesting examples of inductive coupling are presented in reference (62).

The magnetic-coupled torquemeter described in reference (39) has been incorporated into a device at DTMB which they call a "Magni-Torque" pickup. DTMB has also developed a

"Magni-Thrust" pickup for measuring thrust in a rotating shaft without the use of sliprings. These devices, and their principles of operation, are discussed in the transducer section of this paper.

APPENDIX G

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